

Vibration Reduction of Helicopter Blade Using Variable Dampers – a Feasibility Study

By

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A Technical Report submitted to Dr. Chee Tung
US Army/NASA Rotorcraft Division, Ames Research Center
(COTR NASA Grant No. NAG2-1429 entitled A New Semi-Active
Approach for Dynamic Response Modification of Helicopters")

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Abstract

In the report, the investigation of controlling helicopter-blade lead-lag vibration is described. Current practice of adding passive damping may be improved to handle large dynamic range of the blade with several peaks of vibration resonance. To minimize extra-large damping forces that may damage the control system of blade, passive dampers should have relatively small damping coefficients, which in turn limit the effectiveness. By providing variable damping, a much larger damping coefficient to suppress the vibration can be realized. If the damping force reaches the maximum allowed threshold, the damper will be automatically switched into the mode with smaller damping coefficient to maintain near- constant damping force. Furthermore, the proposed control system will also have a fail-safe feature to guarantee the basic performance of a typical passive damper. The proposed control strategy to avoid resonant regions in the frequency domain is to generate variable damping force in combination with the supporting stiffness to manipulate the restoring force and conservative energy of the controlled blade system. Two control algorithms are developed and verified by a prototype variable damper, a digital controller and corresponding algorithms. Primary experiments show good potentials for the proposed variable damper: about 66% and 82% reductions in displacement at 1/3 length and the root of the blade respectively.

Chapter 1

Introduction

In this report, we first review the state-of-the-art and state-of-the-practice technologies for vibration reduction in helicopter blades, followed by details and research plan for conducting a study on vibration reduction for helicopter blades. In addition, we present the major research part; finally, we discuss briefly the future research.

The proposed research consists of four phases:

- (1) Documenting the vibration performance of helicopter blades, reviewing the current vibration reduction methods for helicopter blades, and evaluating criteria and philosophies;
- (2) Identifying major problems of the blade vibrations and discussing possible solutions, new control strategies, and new devices;
- (3) Conducting analytical and limited laboratory studies to develop an improved understanding of helicopter blades under vibration, designing semi-active devices and the corresponding control system design, testing the new control strategies, validating them by experimental investigation, and developing the analytical model for the damper and the system;
- (4) Developing recommendations for improved vibration reduction devices and control system design for helicopter blades.

Task 1 – Conduct Literature Review: Vibration Performance of Helicopter Blades

In this task, we conduct a comprehensive literature review to assess the state-of-the-art in vibration control of helicopter blades. Then we identify and evaluate current published literature from appropriate sources about the vibration performance of helicopters.

Task 2 – Identify of the Important and Key Issues

Subtask 2.1: Identify the Important Issues

Based on the literature review (Task 1), we identify several important issues requiring further research and development. However, because of the limited funding available for this project, addressing all the issues is unlikely under this research. Among the issues to be reviewed are to:

- (1) Clarify and verify the vibration sources in rotorcraft;
- (2) Examine methodologies and mechanism of the vibration reduction in rotorcraft, review existing and anticipated vibration control technologies to determine what techniques are currently available for reducing vibration, and what new technologies are being developed;
- (3) Observe advantages and problems in current vibration reduction methods in rotorcraft;
- (4) Evaluate the performance of dampers (passive, semi-active, and active) the reduce vibration in rotorcraft;
- (5) Assess the possibilities of applying the semi-active control concept in rotorcraft;
- (6) Implement a semi-active control system in rotorcraft, and improve the state-of-the-practice damper design and manufacture;
- (7) Improve control strategies in rotorcraft.

We review and discuss these and other issues identified during Task 1 and recommend further research needs.

Subtask 2.2: Prioritize and Address Key Issues

Based on the above discussion, we identified a number of issues potentially important for developing improved models, and as well as guidelines and criteria for improved design, evaluation, and vibration reduction of helicopter blades.

We ranked and prioritized the final list of important issues as follows.

1. Experimental setup for vibration testing of blades
2. Identification and evaluation of vibration phenomena for the blade, including mode shapes, damping information, and frequencies
3. Design and manufacture of the new dampers
4. Development of control strategies in a semi-active control system
5. Development of experimentally based dynamic models for the blade
6. Validation and updating of the analytical simulation dynamic model, including the finite element model.

Task 3 – Perform Analytical and Experimental Investigations

We propose to investigate the dynamic analysis of helicopter blades using a dual experimental-analytical approach.

Subtask 3.1 – Experimental Investigation – The purpose of this subtask is to set up the experiment to identify the vibration characteristics of helicopter blades, such as the dynamic behaviors, modal densities, influence of boundary conditions, etc. To simulate the performance of helicopter blades in “real” world, an experiment has been set up carefully in the Dynamic Lab, SUNY at Buffalo. The details will be shown later in this report. After accelerations are measured with accelerometers at important points throughout the blade under test, the displacements of the blade are calculated. Accelerometers have measurement noise at the low frequency. Therefore, to avoid errors in collecting the acceleration data and verifying the data and displacement calculation from accelerometers, a LVDT (Linear Variable Differential Transformer) is installed to measure the lead-lag displacement directly. New software has been developed to measure the input and response signals to produce the desired transfer function, and to instantaneously present the vibration characteristics of the blade in the time domain and the frequency domain.

Subtask 3.2 – Identification and evaluation of the vibration phenomena for the blade –

The experiment on the bare blade is conducted, and dynamic characteristics of the helicopter blade are obtained. Based on the information of the mode shapes, damping, and frequencies, etc, we design the semi-active control device.

Subtask 3.3 – Design, manufacture and evaluation of variable dampers – we manufacture and test the new type of the dampers. The vibration phenomena of the blade equipped with the control devices are defined and analyzed.

Subtask 3.4 – Development of the experimental dynamic models for the blade –The experimental dynamic models is generalized in terms of the major control parameters. With a completely calibrated model, we characterize the vibration reduction strategies.

Subtask 3.5 – Validation of the analytical simulation dynamic model – structural identification is applied in this project: a finite element simulation model is developed, and updated by experiment data.

Task 4 – Document Research Results and Prepare and Submit Final Report

Progress report is submitted at the end of the contract. The report contains a description of all research conducted, and discusses the analytical and laboratory studies.

In this progress report, part 1 introduces the research tasks and organization of this report. Part 2 presents the research literature review on vibration reduction in helicopter blades, part 3 proposes our research study on control devices and control strategies, and part 4 shows the design and manufacture of the dampers. In part 5, we present the experimental investigation on the blade behavior with/without the damper. We present several conclusions and discussions based on the experiment investigation, briefly discuss future research work.

Chapter 2

Literature Review

2.1 Introduction

The high level of vibration in rotorcraft causes problems, such as structural fatigue, pilot fatigue, reduced rotorcraft stability and reliability, and increased costs of development and maintenance. Also, the noise caused by vibration affects environmental acceptance.

One of the primary goals for vibration reduction of the rotorcraft is to control vibration within acceptable levels using appropriate means with minimum weight penalty. The barriers to vibration control are accurate prediction methodology and efficient vibration reduction systems. The blade normally has three motions: inplane (lead-lag), out-of-plane (flap), and torsion. Accurately predicting of the lag response is essential because this response is a large contributor to the loads in the blade system. On the other hand, in modern blade design, inplane natural frequency is lower than the rotor speed. This type of rotor system, so-called soft-inplane rotor system, has several advantages, such as lower vibration and hub loads, simpler hub system, and lower maintenance cost. However, the inplane bending mode, or lead-lag mode, of the blade tends to be very lightly damped, requiring a lead-lag damper to dissipate excessive lag motion. Normally, the lag mode natural frequency is around $0.2 \sim 5/\text{rev}$; that is, about $2 \sim 30 \text{ Hz}$.

In the following sections, the vibration sources and current vibration reduction methods are reviewed and discussed.

2.2 Vibration Sources

1. Aerodynamic loading on the blades and resonance amplification of the system

Because of the changes of speed and the angle of attack, alternating air forces act on the blades once per revolution and at multiples of this frequency. If the frequency of the transmitted force is near the resonance frequency of the system, a large amplification of the force and amplitude of the tip motion may cause serious damage may happen to rotorcraft.

2. Vibration caused by some other phenomena, such as blade-vortex interaction

The blade vortex interaction (BVI) causes increased vibration. Because of the interaction between the blade and the tip vortex shed from the preceding blade, BVI consists of vortex-induced loading. At the same time, BVI contains significant low frequency unsteady aerodynamics, which causes substantial increases in low frequency noise content.

3. Aeromechanical instabilities in ground resonance and air resonance

In ground resonance, the force associated with the lag motion reacts with the mode landing gear, causing a huge hub motion. The motion of the hub further excites the lag motion. Air resonance is a coupling of the rotor lag mode with a fuselage mode while the helicopter is in high-speed forward flight. Since the lag mode is only lightly damped, the instabilities can cause blade damage, and even system failure.

As a brief summary, we see the following:

- We may determine the vibration resonances ahead of time, so that we can use the data as a reliable and easy way for vibration active and semi-active control;
- The ratios of maximum over minimum amplitudes as well as highest over lowest frequencies of the blade vibrations render a large dynamic range;

- One of the key issues is that we need large damping to the blade system. Usually the original system possesses only little damping. However, current state-of-the-practice techniques do not provide sufficient damping because of the large dynamic range.

2.3 Vibration Reduction

2.3.1 Passive and Active Materials

Modification of structural properties has included increasing the damping and stiffness. Damping can be increased with constrained-layer damping, damped-link, and interface damping. Each of these treatments has demonstrated the ability to improve the dynamic response of the structure. However, this response is limited to specific locations, modes, or in specific operating conditions. Stiffness can be increased by increasing skin or spar thickness or applying reinforcing members such as brackets, cleats, or doublers. By modifying the stiffness, we can change the vibration characteristic of the blade.

The structures with piezoelectric layers have been investigated. Some research on piezoelectric actuators showed that the piezoelectric actuators do not depend upon the flow characteristics. The most active materials have employed either piezoelectrically actuated flaps at specific locations along the blade, or piezoelectric materials distributed along the blade to directly control deformations in the host blade. The primary constraint in both approaches is the need to obtain high piezoelectric forces and displacements with minimum weight penalty. Since these approaches must be designed to fit within the geometric confines of the blade, they can cause the difficult installation and the possibility of system failure. Some studies indicate the failure during the tests. Also, it has been proved that direct control of blade twisting using embedded piezoelectric is difficult to implement because of the high stiffness of rotor blades in torsion and restrictions in the bandwidth capabilities of available active materials, and while conventional approaches with passive damping materials minimize structural vibrations, piezoelectric layers may add substantial amounts of weight to the structure.

Modern “smart” materials provide an opportunity for on-blade active control, possibly for reduced weight and power. Shape Memory Alloy (SMA) can be trained to remember a specific shape, which it changes back to when heated by an electric current. By arranging the smart materials, some parts of the blade can be changed and moved. Some researchers suggested that SMA can be used as active support struts. However, this method will make the blade structure more complicated. Also, integrating of the smart material and controlled structure is always a problem. The decision on how to optimize the blade shape and how to achieve the target shape will be difficult to make. In addition, SMA consumes more power than piezoelectric layer.

Many researchers have begun to explore the application of ACL (Active Constrained Layer) [17]. ACL consist a layer of high damping viscoelastic material between a base structure and a constraining layer. Nath, etc investigated the feasibility of ACLD (Active Constrained Layer Damping) [26]. They concluded that ACLD was a likely candidate for increasing the damping of the flexbeam. Recently, some researchers studied the application of EACL (Enhanced Active Constrained Layer) damping treatments on the flexbeam [1].

2.3.2 Passive Control Methods

To reduce the vibration level, current helicopters typically employ passive vibration isolation and absorption. Passive control devices, which do not require a power source and can store or dissipate the energy in the system, are inexpensive. However, these devices are heavy, and have various other limitations, such as constant hub loading and constant control parameter.

Passive techniques include optimized rotor blade geometry, tip speed reduction, increased number of blades, and passive dampers. Rotor blade geometry characteristics such as tip shape, thickness, airfoil shape, and blade number significantly impact the vibration level of the blade.

It is crucial to ensure adequate lag-damping to alleviate vibration in ground-resonance and air-resonance instability. Normally, passive lag dampers are implemented to add the lag damping. However, passive lag dampers have drawbacks, such as high damper/flexbeam loads, hub complexity, and constant control parameters.

Elastomeric lead-lag dampers have been used since 1970. These dampers, which dissipate energy by shear motion of the rubber materials, have low weight, no moving parts and no leakage problems [7]. They can reduce the cost of maintenance and the complexity of the hub system. However, elastomeric shear dampers exhibit undesirable nonlinear behavior that is highly dependent on frequency, displacement amplitude, temperature, and loading path. Studies reveal that nonlinearities in the stiffness and loss factor of the elastomeric damper cause the limit cycle oscillation [10]. With the small displacement near the root of the blade, stiffness is high and the loss factor is low; therefore, the damping available for the lag mode is low, and instability occurs. Also, the nonlinearity of the elastomeric material brings the difficulty in theoretical analysis and practical application. The lag motion in helicopter rotors occurs at two frequencies: the lead-lag frequency and the 1/rev frequency. Many studies show that under the dual frequency excitations, elastomeric damper efficiency has been significantly reduced [10, 27]. Elastomeric dampers use shear motion to generate the lag damping force, limiting the placement of the elastomeric damper in the hub zone. The way elastomeric lag dampers are installed in the Boeing 360 rotor hub indicates smaller motion for the elastomeric dampers than that of the hydraulic damper, limiting the capability of elastomeric dampers.

A Fluidlastic snubber/damper is similar to an elastomeric damper except that it includes a chamber which is filled with silicone fluid to provide lag damping [22]. The fluid adds viscosity to the energy-dissipating component of the damper. The advantage of the Fluidlastic damper is that its stiffness is nearly linear, which can eliminate the limit cycle instability.

So far, many studies developed the nonlinear elastomeric damper model as those models established by Gandhi and Chopra [10]. However, few models are associated with the real elastomeric damper.

In past years, effort has focused on design and development of bearingless main rotors for helicopter. The flap hinges, lag hinges, and pitch bearing are being replaced by flexbeams. The flexbeams carry elastic bending and torsion to carry out the flap, lag, and pitch movement. However, a lag damper is still needed to increase lag damping. Yet no clear statement on what is the optimal damping has been addressed. In fact, in many other areas, such as submarine, bridge, and automobile, etc, both experience and theoretical analysis told us that when we need damping, for example, at the resonant region, the larger the damping, the better the result. However, over-large damping may cause two problems. First, because the stiffness of the blade and the fixture are not infinite, large damping tends to become visco-elastic. In other words, when the damping coefficient or its equivalent, becomes larger and larger, the entire system tends to deform and reform. Energy will be stored and released. Such elastic behavior will decrease, not increase the equivalent damping ratio eventually. Secondly, over-large damping may generate extra damping force that may damage the blade-damper-fixture system. Large damping does not necessarily mean heavy damper. If the above-mentioned two conditions allowed, however, increasing damping would be a good strategy for blade vibration control.

2.3.3 Active Control Methods

Active rotor control technologies vary the configuration of the blade or rotor during operation. Active rotor control techniques include higher harmonic control, individual blade control, active trailing edge flap, etc.

Unlike a passive damper, which can only dissipate energy, the active control is able to both add and dissipate energy from the system. The active control scheme is based on the idea that in a linear system, one can superimpose two independent response quantities such that the total response is zero. When applying this scheme to the helicopter vibration

reduction problem, the fuselage at selected locations is excited by controlled forcing, such that the combined response of the fuselage will be minimized.

The primary disadvantage of active control system is that it typically has large power requirements, which adds weight in the rotorcraft. Also, they can have unacceptable failure modes; actuator failure would leave the rotorcraft system undamped. Active systems in general are more costly, more complex, and may bring external energy into the system, making which potentially unstable and therefore less reliable than passive systems.

Research on swash-plate mechanisms resulted in development of HHC [23, 29]. For higher-harmonic control, additional (higher) harmonics are added to the normal 1/rev cyclic input. Selecting the appropriate harmonic and phase angle can change the loading schedule and to reduce vibration level. HHC can reduce helicopter vibration level and rotor noise levels, improving helicopter performance. Conventional HHC achieves higher harmonic actuation of the swash-plate with considerable power penalty. Also, experiments have shown that minimum-noise HHC schedules create high vibration levels and vice versa.

While HHC is limited to having all blades follow the same higher-harmonic pitch schedules, IBC (Individual Blade Control) allows each blade to follow its own pitch schedule [14, 30]. Theoretically, IBC can provide nonharmonic pitch input; therefore, it has much more flexibility than HHC. However, since those methods try to suppress vibration by oscillating the blade at higher harmonics of the rotor speed, they will generate unsteady forces to counteract the effects of air loads. These devices result in weight penalty, complexity of control systems, and higher maintenance cost.

In trailing-edge flaps, a trailing edge flap at the outer part of the rotor blade is considered [24]. A piezoelectrically driven discrete flap generates additional aerodynamic loads to reduce the vibration level at the rotor hub. Because of their potential as lightweight, high

bandwidth actuators with low power requirements, trailing edge flaps can be considered a good control method.

Smart control systems are being developed to detect and reduce vibration. A variety of sensors are used to continually feedback the behavior of the system and the information on vibration levels to a control system. The actuators can then add the forces accordingly to reduce the vibration and noise level.

2.3.4 Semi-active Control Methods

Since an active control system requires an external energy source to power an actuator that controls the structure, and passive dampers have the constant control parameters, the semi-active control system can combine the advantages of both control systems. The semi-active system uses external power only to adjust the damping and stiffness levels. The controller determines the level of damping based on a control strategy, and automatically adjusts the damper to achieve that damping. It needs less power source and has variable control parameters.

Recently, controllable fluid dampers, such as an MR damper, are an attractive choice for the lag damper [9, 13]. MR fluid consists of viscous-elastic plastic fluid containing micron-sized particles. When the fluid is subjected to the magnetic field, the particles can create chain structures parallel to the direction of the applied field. This effect is reversible and fast in the order of milliseconds. An MR damper has many attractive features, such as high yield strength, stable hysteretic behavior over a broad temperature range, and low plastic viscosity. One of the advantages of an MR damper is that it can continuously adjust the damping coefficient. However, current control theory fails to prove the necessity of such adjustment. Normally, when we need damping, the larger the damping, the better the vibration reduction we can achieve. In other words, in most situations, if not all, we need sufficiently large but not continuously adjustable damping. When damping force becomes less important: for example, when the vibration is far from the resonance region, damping is adjusted to be small to avoid extra wear-off. In fact, we

can simply switch off the damper. Still, no continuous adjustable damping is needed (Constantinou and Symans). MR dampers have intrinsic nonlinear properties, which may be undesirable. Also, it moderately depends on the temperature and strongly on the applied field strength, lag motion amplitude, and excitation frequency. Leaking fluid in MR dampers can cause serious physical damage to the human being and to the environment.

The following table summaries some smart materials, including ER fluid.

Material	Force	Motion	Frequency	Power	Pros	Cons
Piezoelectric	Med	Low	High	Low	Sensor/ actuator	Less effective
Magnetostrictive	Low	Med	High	Low	Low power	Temp. sensitive
SMA	High	High	Low	Med	Large force	Low frequency
ER Fluids	Low	Med	Med-Low	Med	Lower cost	Passive force

Although variable damping is attractive in principle, it has problems with respect to varying the damping by using admissible control law. Many researchers suspect its effectiveness. In most cases, the larger the damping is, the smaller the vibration level will be. Therefore, the damping in a structure should not be varied but should be kept at the highest value possible [4].

The other way to achieve semi-active control is through variable stiffness approach. Kobori et al. [15] proposed and installed an active variable stiffness (AVS) system on a full-scale building. Bobrow et al. [2] proposed an active truss element. This truss element can vary its stiffness. Its control diagram is shown in Fig 2.1. The conventional control diagram is shown in Fig 2.2 for purpose of comparison.

The corresponding control law is to switch the auxiliary stiffness at each point when the vibration displacement reaches its peak value. In this way, the auxiliary stiffness can absorb the maximum deformation energy, then dump the energy out when it is switched

off. The power needed is only to change the configuration of the hydraulic valve. This control system is believed to be stable in the sense of energy dissipation by nature. However, this statement is only valid for a free decay system, where no energy is added after the initial stage. Earthquake excitation is a forced vibration; the control law (the resetting logic) given by Borrow is not necessarily stable, in the sense of whether the vibration level is reduced or enlarged. The reason is the possibility of overdraft since the control law could not incorporate the input effect, especially for random inputs.

Lee et al. [16] had overcome the above difficulty and developed a new structural control strategy, Real-time Structural Parameter Modification (RSPM). The basic idea of RSPM is to vary the dynamic stiffness of a structure in real time. The principle of changing the dynamic stiffness is to minimize the conservative energy of the structure [18, 19]. The control algorithms consist of four levels of hierarchical controls. The device to realize the control action is called functional switch, as shown in the control diagram Fig.2.3.

The power out from a controller is used only to change the configuration of the functional switch, so that this control system is stable by nature. The possible overdraft, however, can be eliminated by incorporating higher-level control algorithm.

Fig. 2.2, the diagram of typical active control, shows that from point A to B, then through the feedback path to A again, forms a complete loop, which is the essence of the active control. On the other hand, the switching type of "semi-active" control does not possess a complete loop. The break of the loop is caused by the device. The input to a typical device is often an electric signal determined from the feedback based on the structural responses and issued by the controller. The electric signal commands the functional switch to open/close or to vary the orifice. Since the device is not an actuator, no active force can be generated by the device. Therefore, the device cannot actively affect the behavior of the structure. In other words, the output of the device cannot be controlled by the electric signal only. Quite often, it will not have any direct relation with the signal at all. Therefore, the enclosure of the control loop does not exist. In this situation, it can be very difficult to impose the active control law. In this sense, the "semi-active" control

does not behave like an active control, but close to passive control, since the force that "generated" by the control device is passive in nature.

Although the response of a vibration system is not directly related to the status of a control signal, it is directly related to the physical parameters and external excitations. Changing the physical parameters (mass, damping and stiffness) will result in changing the system response. Obviously, properly and adaptively changing the physical parameters of the dynamic system in real time will result in reducing the system's responses. The RSPM actions refer to those actions that adaptively change the structural parameters, similar to the reflections of a living system to external disturbances. These actions are generated by combined operations of relevant structural elements and strategically installed functional switches. The functional switch is an important component in an RSPM control system. This special hydraulic device that can provide different functions. It includes a rigid link to "switch on" (or connect) an additional stiffness to the structure. It also can provide a "free condition" (or disconnect the stiffness). It can, in the third case, modulate the damping coefficient to achieve a specified damping. A separate publication is dedicated to the dynamic responses of the functional switches.

Decision-making depends not only on the responses of the structure, but also on the state of the excitation force. With the RSPM actions, the structure performs an adaptive function against external excitations through a certain modification law, so that the vibration is reduced. There are no external control forces in an RSPM control system although the function switching provides a mathematical equivalent control force. This is an important and extremely desirable feature in earthquake engineering applications. RSPM control, or variable passive control, intends to express the nature of the system having the ability of an active control system as traditionally defined but without the demand for instant availability of power to operate large actuator in order to deliver counter forces that must be applied by external means.

Stiffness modification is often used in most structural control applications because of its

effectiveness and its implemental ability. This report explains the essence of RSPM, which modifies stiffness as well as damping and mass, with major emphasis on variable damping and the dynamic characteristics of switching. In order to unveil the special damping mechanism, an RSPM scheme with stiffness modification is examined. The model of such a system is given in Fig. 2.4.

In Fig. 2.4, the entire mass M of a structure is lumped in the rigid block, which is constrained by rollers so that it can move only in simple translation. Thus, the single displacement coordinate $x(t)$ suffices to define its position. The permanent and switched stiffness are represented by the weightless spring K and ΔK , respectively. The damping is represented by dashpot C . FS is the functional switch that realize the RSPM actions. The sensors and decision making unit required in RSPM, are not shown or discussed here.

It can be shown theoretically that the proposed method of RSPM for variable passive control can dissipate energy four times that of variable stiffness control. Practically, the amount of energy dissipation will also depend on the reaction time of the functional switch. Experimental data show that, if the natural frequency of a structure is around 2 to 3 Hz, or below that number, the time period of valve opening and force dropping should be no longer than 30 to 40 milliseconds. This number can be easily achieved with available technology. The frequency range is commonly concerned in earthquake engineering. However, the natural frequency of rotor blade can be higher. Thus we need shorter responding time for the blade experiments. So far, the quickest reaction time we can achieve is about 2 to 3 milliseconds by using a very fast solenoid valve. For vibration control on the rotor blade, it can be sufficiently fast and the energy dissipation can be quite close to the theoretical value. As mentioned before, a large amount of energy dissipation with relatively low force is one of the two major goals of this study.

In comparison to typical viscous damping with the same amount of energy dissipation, the required force is in fact slightly higher, not lower than the viscous damping force. However, since the viscous damping force is by nature a damping force that is proportional to velocity, that force can contribute little to the force proportional to

stiffness. In many cases, such a spring force is needed to increase the overall dynamic stiffness. In this regard, RSPM will improve greatly since the control force generated by functional switches is in nature a spring force. Figure 2.5 shows conceptually that when the spring force controlled by functional switches is separately considered, the RSPM scheme can provide the maximum energy dissipation with given bounded forces and displacements. In a later section, the corresponding increase of dynamic stiffness by using the RSPM scheme will be given.

As mentioned before, if the control scheme is carried out only by the maximum energy dissipation, shown in Fig 2.5, the total control can be unstable because of the phenomenon of overshoot. RSPM thus will include a hierarchical control measure, using a multi-ranked control loop. The first loop will be designed to achieve the maximum energy dissipation possible, using a push-pull functional switch pair. The second loop, which overrides the first loop, is designed to avoid the overshoot.

If more than one pair of functional switches are used, each pair can be treated as a local control; the total control of all sets of functional switches is said to be global. The third loop is the global loop that overrides the first two loops to achieve the optimal dynamic stiffness for the entire structure. The fourth loop is used to guarantee the fail-safe function. Details of this control principle are given in the separate publications [18, 19].

More details about the semi-active control devices and theories in this study will be discussed later in this report.

2.4 Theoretical Analysis

Modal analysis is a method to construct a mathematical model of the vibration characteristics of the system based on the collected experimental data. In the modal analysis, the modal model must discover three main properties: frequency (resonance frequency), damping (damping ratio), and residue (mode shape, magnitude, and phase). The information about magnitude and phase of the system can be obtained by calculating

the residue. Every mode shape is associated with the specific deflection of the system. The frequency response functions (FRF) can be used to obtain those modal properties.

A process so-called curve-fitting is needed to estimate the modal properties to generate an analytical function. In MDOF curve-fitting techniques, four methods can be used to estimate modal properties: complex exponential, direct parameter, poly-reference, and orthogonal poly-reference. The complex exponential uses a single response function from one single reference location. The direct parameter uses multiple response function from one single reference location. The poly-reference uses multiple response function from one or more reference locations. The modal properties will show the overall influence of all the modes in the frequency domain. The orthogonal poly-reference uses multiple response functions for different reference location in the frequency domain. A software package has been developed to do the curve-fitting in this project.

2.5 Summary

The problem of blade vibration is not adequately solved to our satisfaction. The performances of various control methods do not sufficiently reduce the vibration level. Yet relatively larger power, higher voltage, and heavier weight could still be further reduced. The resonance problem still exhibits. Large dynamic instability is observed in the system, even with the vibration reduction devices. Usually optimal damping for given structures is the largest damping possible to be added. The reason why conventional design cannot efficiently provide large damping is partially that current design standard is based on worst scenario design for the safety issues. Due to large dynamic range of blade vibration, most of the control devices cannot achieve the control goals.

In our research, our goals to reduce the vibration level are: First, we should avoid the resonance by verifying stiffness; Second, adjustable damping makes it on-line-optimal to reduce the blade; Third, the control devices should have a fail-safe mode with default damping close to conventionally designed valve. More details will be given in the next part.

Chapter 3

Research on Semi-active Control Device

3.1 Introduction

In this chapter, we focus on the issue of adding damping to the blade systems, the problem identified in the last chapter.

Before we begin to design the control system to reduce the vibration in rotorcraft, we should keep in mind that the control system:

1. Should not cause too much weight penalty
2. Should not add the cost much for produce and maintain
3. Should not add the complexity of the helicopter system/control system
4. Should have high level of stability and robustness
5. Should have a safe function in case the control device itself is malfunction.

By examining the state-of-the-practice for vibration control on helicopter rotor blades, it became clear that the problem is not adequately solved to our satisfaction. The performances of various controls do not sufficiently reduce the vibration level. This study is motivated to examine the blade vibration problem based on the experience of using semi-active control to reduce seismic vibration.

3.2 Vibration Reduction Devices

Damper behavior can strongly influence the behavior of the rotors. Because the stability of the rotorcraft, the blade load, and the in-plane motion are coupled with damper properties, the damper should be carefully designed to reduce the vibration level of the rotorcraft.

At least two important issues must be addressed in semi-active control: control principles and control devices. Two types of hydraulic variable damping devices have been

proposed by many researchers. First, Feng and Shinozuka [7] presented the varying orifice. Mizuno [25] proposed an adjustable hydraulic damper that controls the damping force by means of a flow control valve between two chambers of the hydraulic cylinder. Patten and Sack [28] proposed a semi-active vibration damper, also adopting the variable-orifice concept by using Lyapunov functions. Constantinou and Symans [4] have studied another kind of variable-orifice damper, which regulates the secondary orifice. Second, varying apparent viscosity of the fluid can accomplish variable damper. Electro-Rheological (ER) fluid and Magnetorheological (MR) fluid can be used as this type of damping material.

One way to achieve the semi-active control is to use a hydraulic damping mechanism for variable damping. In this study, variable damper is proposed. By controlling the fluid flow path, the different damping force can be provided. One fluid flow path is through the small orifice, which can create the larger damping force. The other fluid flow path is through a large channel, whose diameter is much larger than that of the small orifice. So, in this state, a relatively smaller damping force is created. Therefore, the control device can provide the variable damping to the structure, even when the structure has only small amplitude of displacement.

The hydraulic resistance through an orifice is a function of the orifice flow area, the apparent viscosity of the fluid goes by, and some other factors. The damping force is the result of viscous friction arising from the passage of the working fluid through an orifice. We will present the details in the mathematics model of the damper.

The schematic of the damper is detailed in the Appendix.

3.3 Control Strategies

From the Coleman diagram for frequencies (ground resonance analysis) in Figure 3.1 [3], we know that in the instability zone, whenever there is coupling of the modes, one mode becomes unstable and the other mode becomes even more stable. Also, one mode will be

more damped than the other one. As we know, normally the one that is less damped becomes unstable. In a helicopter, the lag mode has lower frequency and is less damped. Typically, the lag mode becomes unstable.

From the Coleman diagram for damping (ground resonance analysis) in Figure 3.2, we know that with the increased rotor speed starting from the helicopter run-up from the ground, the rotor must go into and out of the instability zones. When the rotorcraft is in the instability zones, a large damping force is needed to reduce the vibration. When the rotorcraft is out of those instability zones, a small damping force is needed to keep the hub loads and fatigue loads small. Therefore, damping augmentation is required only in certain flight situations, where a potential for instabilities occurs, such as ground resonance and air resonance. In other flight regime, the strong damping force provided by the passive damper or elastomeric damper to alleviate ground resonance and air resonance produces excessively large periodic damper loads (due to periodic flap/lag/torsion motions of the blade.). These loads increases fatigue loads and maintenance cost, thereby reducing service life.

In some situations, the force in lead-lag direction is too high, if the damper keeps providing high damping force, it may cause damper damage, and even the whole system failure. Therefore, the variable damper, which can provide different damping force for the different flight conditions, and increase damping when critical and provide less damping if not necessary, would be a better candidate for the lag damper in the rotor system. However, we must keep in mind that the support stiffness provided by the connection between the damper and the blade is not infinite in the real world. Therefore, if the rotorcraft is out of the resonance zone, and if the damper provides less damping force, the total force provided by the damper and the connection stiffness is reduced, which can cause the natural frequency of the hub to shift to the lower frequency. This lower frequency may couple with some of the low natural frequency of the fuselage, causing the resonance. Therefore, when we decide to take advantage of the out-of-resonance-zone flight and provide less damping force, we must design the control algorithm carefully to avoid the system's falling back into the resonance zone.

As we know, a dynamic system has three parameters, which can be used to change the dynamic behavior of the system: mass, damping, and stiffness. In the helicopter blade, it is not easy, if not impossible, to change the mass. Therefore, besides the damping, we can change the stiffness to reduce the vibration level of the blade. The stiffness will shift the natural frequency in the system. In this study, we propose to use the support stiffness to achieve our goal. The support stiffness exists in the connection between the damper and the blade.

We use the following control strategies in this study:

First, we vary the damping coefficient to keep constant damping-force. When the vibration level is small, a large damping coefficient is used to ensure sufficient damping effectively suppressing the vibration. When the vibration level is higher than a given threshold, the damping coefficient is shifted to be small to guarantee that the combined force does not to exceed the allowed level, and vice versa. .

Second, we change the supporting stiffness to avoid any possible resonance. That is, when the blades start to rotate, the stiffness is high to achieve higher natural frequency. When the rotating speed becomes higher and close to the natural frequency, the stiffness is switched to be low to have another natural frequency, but lower than the driving frequency. This portion of control is not completely tested.

Third, a fail-safe mode is considered as the default setup.

3.4 Summary

In this chapter, we discuss the new control strategies for the newly-designed variable damper. Theoretically, we verify the concepts of the control strategies. In the next chapter, we discuss the experimental investigation on the damper and the system. Based on the test data and results, we can validate the damper and the control strategies from the stand-point of experiment.

Chapter 4

Design and Fabrication of the Dampers

In this chapter, we study several dampers. First, we use friction damper with fixed levels of forcing functions as a tool to design variable dampers. We then use an in-house made MR damper to search the limit of the effectiveness of variable damping as well as the performance of the MR damper. We finally design and fabricate the proposed variable hydraulic damper accordingly.

4.1 Friction Damper

If a friction damper is designed under an idealized condition; i.e. steady state vibrations, it can provide the optimal damping. The reason is that the damping constitutions, namely the relationship between damping force and damper displacement, is close to rectangular-shaped energy dissipation. Therefore, an adjustable friction damper is first used to determine the proper parameter of variable damper. Basically, it can:

- (1) Verify the possibilities for applying the passive dampers.
- (2) Discover the allowed blade's the maximum damping force under sweep sine input, upon which the designs of the semi-active damper are based.
- (3) Discover the dynamic characteristic of the blade with a friction damper, the result can be compared with the results from semi-active devices.

The details are shown in Figure 4.1. During the tests, the friction damper can be in three states: completely fixed, loosened, and completely off. From the tests of friction dampers, we know that the best design damping force for the semi-active damper is about 1000 lb.

4.2 MR damper

The MR damper is designed to determine:

- (1) Possible application of the MR damper in reduction of the vibration for blade
- (2) Dynamic characteristic of the MR damper
- (3) Dynamic characteristic of the blade with MR damper
- (4) Verification of the control system for the MR damper used in the blade system
- (5) Comparison the experimental results with the results in the friction damper and variable damper.

It is also used to study the possible limits of vibration reductions by adding damping to the blade systems. It is found that the MR damper does vary the damper coefficient continuously. However, two major problems present: first, its control range is relatively small. By means of a simplified computer simulation, it is found that our MR damper cannot deal with the large dynamic range of blade vibration sufficiently. Second, the MR fluid we purchased is not quite stable. The viscosity of the fluid varies. Immediately after a thorough shaking, it has the desirable damping force. But the viscosity decreases in minutes. After two hours, the damper loses its performance. We did test the MR damper with an MTS machine and the blade setup. Since the performance is not reliable, we shall not discuss it in the later chapter.

The details of the MR damper are shown in Figure 4.2, Figure 4.3, and Figure 4.4.

4.3 Variable Damper

As mentioned before, the proposed variable damper is designed to achieve two goals. First of all, it adds additional stiffness. This goal can be reached by the characteristics of visco-elastic behavior of the assembly of the damper and its fixture. During the motion of the damper piston, energy is always stored and released by the deformation of the fixture because the stiffness of the fixture is not as infinite. Such energy transformation will actually add certain stiffness to the entire blade-fixture-damper system and will vary the system's natural frequencies. The second goal is to add damping.

To achieve these two goals, the variable damper has two fluid paths. Path I has a larger diameter so that when a certain amount of hydraulic fluid is squeezed through this path, energy is dissipated but not a very large amount. Because Path II has a considerably much smaller diameter, much more damping force can be obtained when the large amount of fluid flows through. In addition, since the damping force becomes larger, it will deform the fixture to achieve additional stiffness. The special fixture is carefully designed and fabricated.

The details are showed in attached drawings in the Appendix.

The prototype semi-active control damper is designed to have the following features:

Stroke: 0.1mm-25.4mm (0.004 inch ~ 1 inch)

Working damping force: 454 kg (1000 lb)

Working frequency: 1 – 50 Hz

Response time: 10 ms

Chapter 5

Experimental Investigation

5.1 Introduction

The experimental investigations are focus on the following aspects:

1. Dynamics characteristics of the new-designed hydraulic damper
2. Dynamics performance of the blade with/without the damper
3. Advantages of the new-designed hydraulic damper
4. Lessons we learn from the experiment investigations

Based on these aspects, we carry out the experiment investigations. We design and conduct the experiment setup, including excitation methods, boundary condition, sensor layout, and setup of the data acquisition system. After we obtain the experimental results, we have the dynamic characteristics of the damper, and the performance of the blade with damper. W can then compare the results, and draw conclusions.

5.2 Excitations

Two input methods are used in the test. One way to excite structures is to use an electro-dynamic shaker. A force sensor attached to the armature driving the test object is used to define input force amplitude while a signal generator controls frequency. The shaker is fixed to the ground, allowing the highest possible excitation levels. Since this type of excitation mechanism is easily controlled in frequency and amplitude, it offers the best overall accuracy. An alternative method is to strike it with a calibrated dynamic impulse hammer. A force sensor mounted in the head of the hammer transforms the input force pulse into an analogous waveform that contains the necessary amplitude and phase information to completely describe the forcing function.

In this project, both excitation methods are used. The shaker used in the test is Model 1068, Vibration Test System. It operates over a frequency range of 0- 5000Hz. The power supply amplifier for the shaker is Techron 5515, Crown International, Inc. The hammer is from PCB. The shaker setup is showed in Figure 5.1.

A new in-house software package is specially designed to generate the required sweep sinusoidal signal and to extract the dynamic characteristic of the blade in the frequency domain. We can specify the frequency ranges, the number of sine dwelling cycles, and the amplitude of output. The output of the accelerometers from the blade is filtered first by correlations integration, allowing us to achieve much higher signal-to-noise ratio and virtually no leakage. If the shaker can be driven with sweep sine signals, it concentrates all the energy in the frequency of the range of interest. Therefore, we can have much more accurate data. Since in the sweep sine signal, we can collect more data when the signal is around the resonance frequencies, and collect fewer data between the resonance frequencies, the data can be collected much more efficiently and accurately. Figure 5.2 shows the screen of in-house software.

This in-house software package can also be used to simulate the aerodynamic behavior of the blade in the flight condition. It can send out the signals from the low driving frequency to the high frequency, then from high frequency back to low frequency, which can simulate the rotorcraft running up in the ground, and going in and out of the resonance zone. At the same time, a function in this software package allows us to set up a switch signal. After the sweeping sinusoidal arrives at the switch frequency, a switch signal (in voltage) will be sent out to the controller. By control algorithm, the controller will send out a control signal to switch the fluid flow state between the small orifice and large channel in the damper. In this way, we can observe the performance of the semi-active damper. The notation “auto sweeping test” is used in this report for this function.

5.3 Boundary Conditions

The boundary condition in vibration analysis directly impacts the modal characteristics of the system. Two types of idealized supports for vibration testing are fixed support and “free” support. In fixed support, there is a constrained attachment to some no-motion bases. “Free” support simulates of the structure floating free in space with air lifting force. Since in the “real” world, the blades are fixed with the hub, so the fixed support system is use in the test. Only lead-lag motion rotation in the large hole of the blade is allowed. The homemade bolt in the large hole of the blade is allowed only ± 0.010 inch. Note that, in the real flight condition, the helicopter blade will have the distributed lifting loads along the blade in flap and lag directions, and a centrifugal force in the axial direction. Because of the limitation of the experimental equipment and lab space, we only use the centralized lifting load in 1/3 length of the blade from the tip. Four rubber jump cords are used as the hanger to lift up the blade. The height of the blade away from the ground can be controlled by the chain, which is connected with the jump cords. No centrifugal force is simulated in this study.

The blade is cantilevered to the steel bases at the end of the blade. The steel bases are fixed on the concrete ground, a five-cell reinforced concrete box girder 40 ft long, 60 ft wide, and 8 ft overall in height. The thickness of the top test floor slab is 18 inches. Tie-down points consist of 4x2½ inch symmetrical holes. Each tie-down point has an axial load allowable capacity of 250 Kips. Figure 5.3 and Figure 5.4 show the details.

To reduce the influence of the shaker on the blade, a stinger between shaker and blade is used and it must be carefully chosen. The stiffness and length are the important parameters. If the stinger is too long, it can vibrate laterally; if too short, the behavior of the shaker and stinger will be added into the blade and mixed with the dynamic behavior of the blade. In this case, the data are not accurate. After several trials, the stinger, whose length is 14.5 inches and diameter is 5/32 inch, is chosen. An aluminum base is used to connect the stinger and the blade. The curve of one side of the aluminum base is fabricated according the shape of the leading edge of the blade. Two aluminum bases,

one for the hammer test, and the other for the shaker test, are glued with the leading edge of the blade.

5.4 Node Layout

The node layout is detailed in Figure 5.5. The placement of the exciters and the sensors are chosen properly to excite all the modes of interest are excited, and unique geometrical description of the mode shapes. Improper node layout may cause wrong dynamic characteristics of the blade. To obtain sufficient accuracy and readability of modal shapes, the blade has been discretized in 14 points, which are also constrained by the number of channels in filter and conditioners.

There are two load cells in Node 9 (Y direction) and Node 2 (X direction). The sensitivity for the load cell in Node 9 is 20 lb/Volt; the one for the load cell in Node 2 is 1000lb/Volt.

5.5 Data Acquisition

The accelerations on the blade have been measured by using a set of PCB accelerometers mounted on a small plastic base. The curves were specially manufactured according to the shape of the leading edge of the blade. The plastic bases are glued on the blade, and the accelerometers are screwed onto the plastic base. The masses of the plastic bases and accelerometers are very small compared to that of the blade.

The linear variable differential transformers (LVDT) are used in displacement measurements. The scale for the LVDT in Node 2 and Node 9 is 0.2 inch/Volt. Low-pass filter is used to avoid signal aliasing.

The in-house software package and Virtual Bench (National Instrument) are used to collect the experimental data. Virtual Bench collects the dynamic response in the time domain, such as acceleration and displacement from accelerometers and LVDT. The in-

house software collects the data in the frequency domain, including magnitude of the vibration, frequency, phase, etc.

A controller system is designed to implement the control strategy according to the response of the blade. This controller system includes a computer with control algorithm, a controller box for receiving the control signals and sending out the signals to the directional control valve in damper, and a power supply for the controller box. The control signal is either 0 volts or 12 volts. Figure 5.6 shows the controller setup.

Figure 5.7 shows the data acquisition system and Figure 5.8 shows the picture.

5.6 Damper Mechanism

Figure 5.9 shows the damper in the blade.

When the power for the directional control valve is on (12 volts), the large fluid flow channel (diameter: 0.386 inch (9.80 mm)) is open and the damping force will be small. When the power is off, the large fluid flow channel is closed and the fluid must flow through the small orifice (current used diameter: 0.037mm), creating a larger damping force. The size of orifice can be chosen according to the damping force and dynamic behavior of the rotor system. Also, the volume control needle valve, which is in the same flow path as the directional control valve, can be used to manually adjust the speed and volume of the fluid flow and create the different damping forces.

For convenience, in this report, “small orifice” and “large channel” are used to describe the two different control states. At the same time, “off” is used for the “small orifice” state in the plots, since the power is off in this state, and “on” is used for the “large channel” state, since the power is on in this state.

The diaphragm can be used to absorb hydraulic shocks, compensate for leakage, and improve the stability of the damper by dampening the flow fluctuations. The check valves allow free flow in one direction while preventing flow in the reverse direction. When the pressure in the cylinder is lower than that in the diaphragm, the fluid can flow from the diaphragm to the system. However, the fluid cannot flow in the reverse direction from the system to the diaphragm by the check valves. If the pressure in the system is too high, the relief valves will open and the fluid can flow to the diaphragm. The relief valves offer stable performance and fast response.

5.7 Experimental Results

5.7.1 Component Test of the Hydraulic Damper

To investigate the characteristics of the hydraulic damper, component tests of the damper are conducted on MTS axial-torsion testing machine. This machine is capable of biaxial testing of specimens. Control modes available are force, strain, and displacement in axial mode. The machine has calibrated ranges of 100, 50, 20, 10 Kips, and ± 5 , ± 2.5 , ± 1 , and ± 0.5 inch axially. In this study, displacement control is used. The displacements are 0.01 inch (0.254 mm), 0.02 inch (0.508 mm), and 0.03 inch (0.762 mm), respectively. The component tests include the performance of the damper in 1 Hz, 2 Hz and 4 Hz frequency sinusoidal inputs.

The force-displacement relations of the damper in the small orifice state and in 1 Hz, 2 Hz and 4 Hz are shown in Figure 5.10, Figure 5.11, and Figure 5.12, respectively. The force-displacement relations in the large channel state and in 1 Hz, 2 Hz, and 4 Hz are shown in Figure 5.13, Figure 5.14, and Figure 5.15, respectively. The upper plot shows the force-displacement relation, the middle plot shows the displacement signal in the time domain, and the lower plot shows the force signal in the time domain.

The comparisons of the force-displacement relations between the small orifice and large channel in each frequency and displacement are shown in Figure 5.16, Figure 5.17, and

Figure 5.18, respectively. Table 5.1 lists the comparison of damping energy ratios. The damping energy is the area that one cyclic force-displacement can cover. These ratios can show us the damping capacities of the damper under the different excitations (different frequency and different amplitude).

Table 5.1 Comparison of Energy Ratio

	Small Orifice Energy (kips·inch)	Large Channel Energy (kips·inch)	Ratio
1 Hz, 0.01 inch	0.0049	0.0033	1.51
1 Hz, 0.02 inch	0.0189	0.0053	3.56
1 Hz, 0.03 inch	0.0487	0.009	5.41
2 Hz, 0.01 inch	0.0088	0.0026	3.34
2 Hz, 0.02 inch	0.0436	0.0057	7.71
2 Hz, 0.03 inch	0.1251	0.0078	16.08
4 Hz, 0.01 inch	0.0195	0.0025	7.84
4 Hz, 0.02 inch	0.1138	0.0051	22.42
4 Hz, 0.03 inch	0.2015	0.0076	26.53

From the test data, we know that the variable damper achieves the control goals and can be installed in the blade for the further test. The ratios are increased with the increasing amplitude. With the different frequency, the small orifice energy has dramatic change, and the large channel energy has little change. Also, based on the damping constitution, we can design the fixture between the damper and the root of the blade.

5.7.2 Helicopter Blade Test

The blade test has three courses. The first is with the large diameter fluid path only, denoted by “large”. The second is with the small diameter fluid path denoted only by “small”. The third is with the variable damper control denoted by “control”.

Figure 5.19 shows the displacement of the blade with the variable damper, which is located in $1/3$ the length of the blade away from the tip. Figure 5.20 shows the displacement of the damper. Figure 5.21 shows the displacement in the location of the accelerometer in the tip, which is calculated from the acceleration data. It is clearly seen that: first, the large channel test and the small orifice test exhibit distinct natural frequencies. During the large channel test, we can visually identify that the peak response resonates at 2.8 Hz. In the small orifice test, the resonance frequency is 6.5 Hz. These figures indicate that our first strategy of varying the stiffness works. Second, from Fig 5.19, we can roughly see that in the large channel test, the system presents smaller damping ratio, which is about 28.6%. Since we use the small fluid path in small orifice test, comparably much higher damping ratio is achieved, which is about 84%. In the control test, we use the computer to identify the threshold of switching frequency and thus apply the controlled damping/restoring forces to the blade. Clearly the amplitude of the controlled response is significantly reduced from the large channel state to the small orifice state. With the control, the peak value of the displacement measured at $1/3$ length of the blade is about 0.21. Without the control, the amplitude is about 0.72. The uncontrolled responses are about 3 to 4 times larger.

The displacements of the blade and the damper in auto sweeping tests are shown in Figure 5.22 and Figure 5.23. The auto sweeping tests also consist of the small orifice test, large channel test, and control test. From these figures, we can see the same reductions on the displacement of the blade and the displacement of the root of the blade.

These facts imply that our second strategy can work with the variable damper. In real helicopters the stiffness of the damper assemblies may be quite different from our test.

Here, we only conduct the proof-of-the-concept test; reasonable questions may arise that in real situations the variable damper may still achieve the additional stiffness to avoid resonance. In the large channel test and the small orifice test, the damping ratios are quite high. In real cases, according to the literature, it will be only 5%. In such a condition, if we can achieve 20% difference in natural frequencies, we can successfully avoid the phenomena of resonances.

It is noted that these large damping ratios, 28.6% and 84% respectively for different modes, are not only contributed by the variable damper, but affected by the aforementioned boundary conditions. Such a boundary condition is not realized in real applications. Instead, in real circumstance, the damping energy dissipating contributed by the blade system is rather small. It is also noted that, if possible, we would rather use smaller damping contributed by the boundary conditions. The smaller that portion of energy dissipation is, the larger difference of the two pre-set damper modes can be achieved. However, in the lab conditions, we have to deal with that large damping, which in fact makes the test less effective.

5.8 Summary and Discussions

- (1) Control strategies review: the damping force can be switched according to the different flight situation, while the stiffness can be switched accordingly:
 - When the vibration of the blade approaches to the resonance region from driving lower frequency, additional stiffness should be switched off. When the vibration of the blade approaches the resonant region from driving higher frequency, additional stiffness should be switched on. In this way, we successfully avoid resonances;
 - In real flying conditions, as the speed of the rotation varies, the vibration of the blade may pass through several resonance regions. The proposed control strategy will apply to all of them if the frequency shift can distinguish the resonance points;

- There are two methods to determine the switching threshold of the frequency. First, the resonance frequencies are determined by the driving frequency, namely, the speed of the blade rotations. Therefore, the information on blade speed will be sent to the controller; with the pre-knowledge of the blade's natural frequency, the switching functions are conducted. Second, we use sensors and the computer to online identify the driving frequency, but identification takes time. We always have room for the calculation because the driving speed does not immediately reach the resonance. We can always have pre-knowledge of the trend of the approaching period.
- When resonance can not be avoided; for example, when two natural frequencies are too close to be switched, we add large damping to the system. Again, the boundary conditions are different from a real flight. The test data show that extremely large damping will not be true in real situations. From the component test, the damper can dissipate efficient energy to achieve large damping ratios of about 84% in our tests. However, as mentioned before, the fixture of the damper as well as the entire elasticity of the blade will not provide efficient stiffness to achieve such a high damping ratio in real conditions. Nevertheless, with this primary testing style, we feel comfortable that our variable damper can contribute larger damping than commonly used ones.
- The reason we are comfortable in providing large damping can be explained again in more detail. In normal situations, when large damping applies, smaller vibration level can be achieved. The resulting damping force roughly a product of the damping coefficient and vibration velocity, will not be very large. Thus, large damping should be used. However, if for some reasons the vibration level starts to grow and finally becomes quite large, the damping force with large damping coefficient will become very large without control efforts. It can be too large to be allowed in the blade-damper-hub system, since an overly large force may damage the blade system; therefore, for safety, the damping coefficient cannot be too large.

Of course, by doing so, we will not have large damping but large vibration. With the variable damper, the initial damping coefficient is designed to be as large as possible, if the stiffness of the fixture and blade can support such a damper. Therefore, in normal conditions, we will have considerably large damping and small vibration. If in the worst scenario, the vibration becomes too large and the damping force may damage the system, the large damping ratio is switched off immediately to prevent the damage.

- (2) The hydraulic damper can provide large damping force even in the small lag motion. In the small orifice state, the displacement of the damper is 0.003 inch (0.076 mm), and the displacement of the blade (1/3 length of the blade away from the tip of the blade) is 0.04 inch (1.016 mm). In the large channel state, the displacement of the damper is 0.0164 inch (0.416 mm), and the displacement of the blade is 0.119 inch (3.023 mm). The displacement of the damper is reduced 82%; the displacement of the blade, 62%.
- (3) The small orifice state has the ability to provide enough damping energy
- (4) The manual needle valve provides a match for the dampers to different rotor systems. Therefore, the dampers have better tracking ability.
- (5) This newly-designed damper is a fail-safe damper. Even if the switch damping function fails, the damper still can act as the traditional passive damper.
- (6) Low power consumption and low voltage (12 Volts) are used to control the variable orifice.

Therefore, this hydraulic damper has improved damper performance. It may potentially reduce maintenance cost, lower hub loading, and achieve less fatigue, and improved stability.

Chapter 6

Future Research

Further studies may include:

1. A fatigue test on the damper

A fatigue test is very important to ensure the reliability of the newly developed damper. Although experience on the hydraulic cylinder, control valves, and accumulator are available among many existing companies, the assembly of the cylinder and the cartridge body in this special design is new. We thus plan to conduct a thorough fatigue test. The test will apply 1000 lb force at 20 Hz driving frequency on the existing damper with/without control commands. External loads, internal pressure, vibration velocity, damper displacement as well as frequency will be recorded. The number of the test cyclic will also be recorded.

2. Further computational simulating to compare with other researchers' work

- Develop mathematical model of damper;
- Optimal design of the controllers to maintain tracking performance, and optimization of the control algorithms;
- Optimization study that evaluates the performance of the damper;
- Finite element-based analytical model of the blade with the hydraulic damper;
- Consideration of nonlinearity (global and local) in the analytical model.

3. Prepare for the flight test, including gathering the information on the helicopter for installation of the damper, adjusting the damper and damper accessories, setting up the new data acquisition system, etc. Online system Identification: the transfer function estimate should be updated fast enough to slowly track variations in the plant

4. Further develop the control software and hardware for practical applications. Include the product of a software package about a modal analysis tool written in MATLAB to exploit frequency domain measurements, to present the dynamic characteristic of the blade, to simulate the blade performance, and to monitor the blade under aerodynamic load. GUIs will be provided for visualization of responses, frequency domain identification, animation of operational deflection of modes shapes, and monitoring the blade damage, etc.

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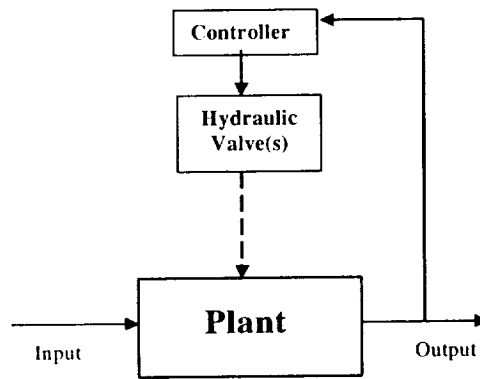


Figure 2.1 Active Control Diagram

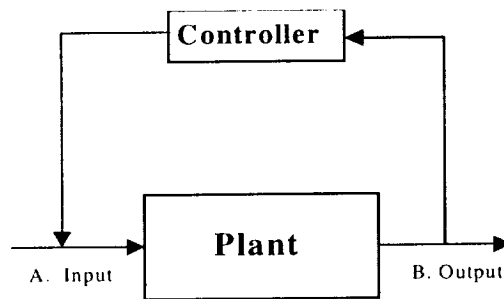


Figure 2.2 Conventional Control Diagram

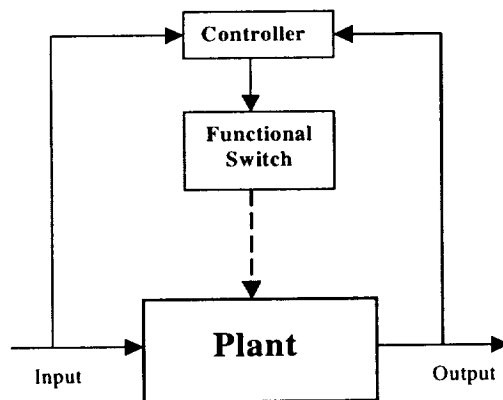


Figure 2.3 RSPM

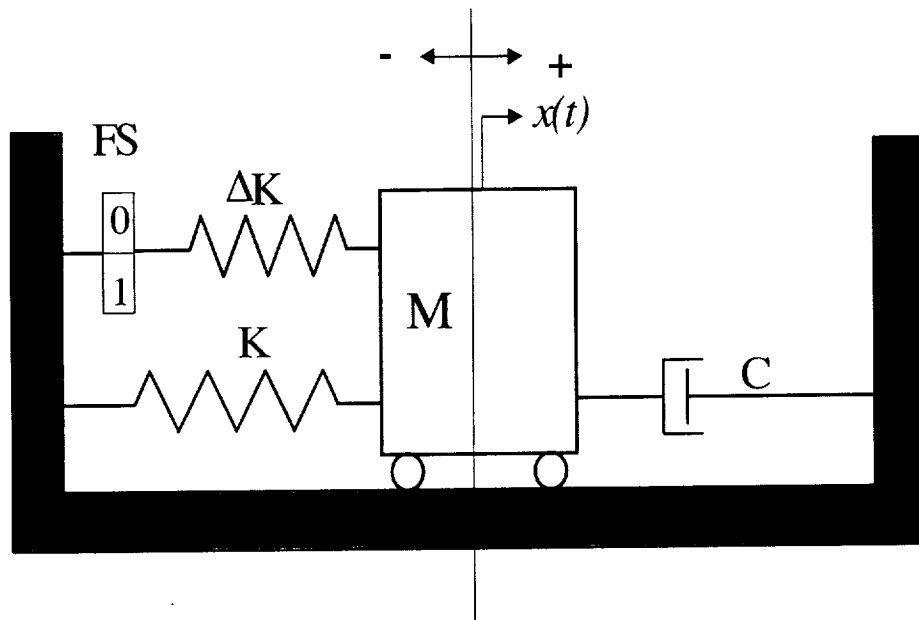


Figure 2.4 Idealized RSPM System

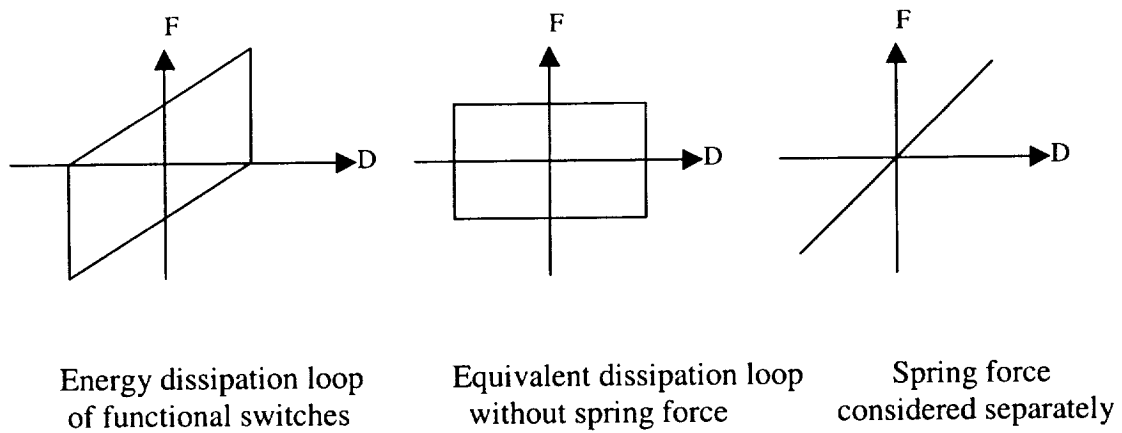


Figure 2.5 Control force and energy dissipation loop

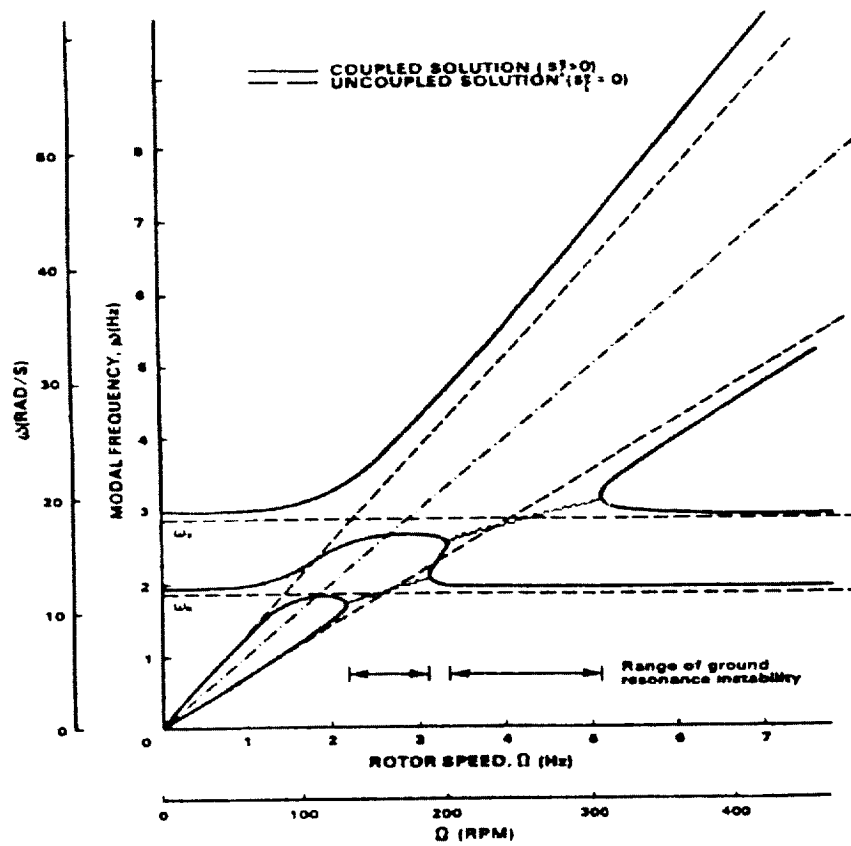


Figure 3.1 Coleman Diagram for Frequencies

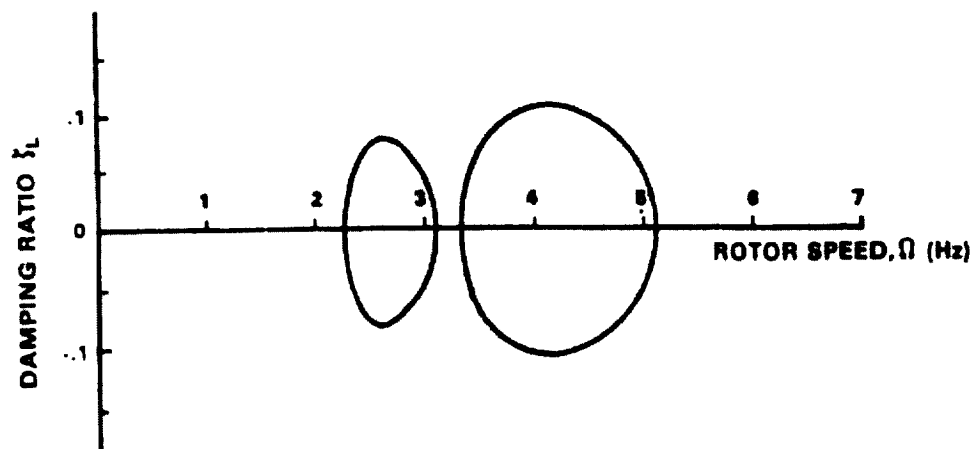


Figure 3.2 Coleman Diagram for Damping

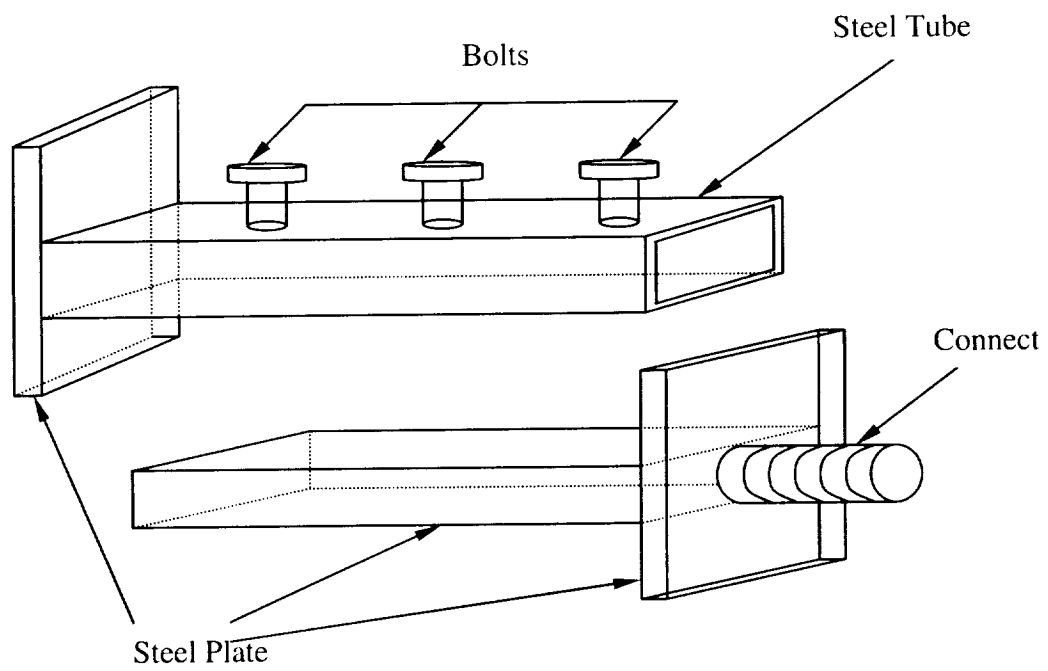


Figure 4.1 Friction Damper

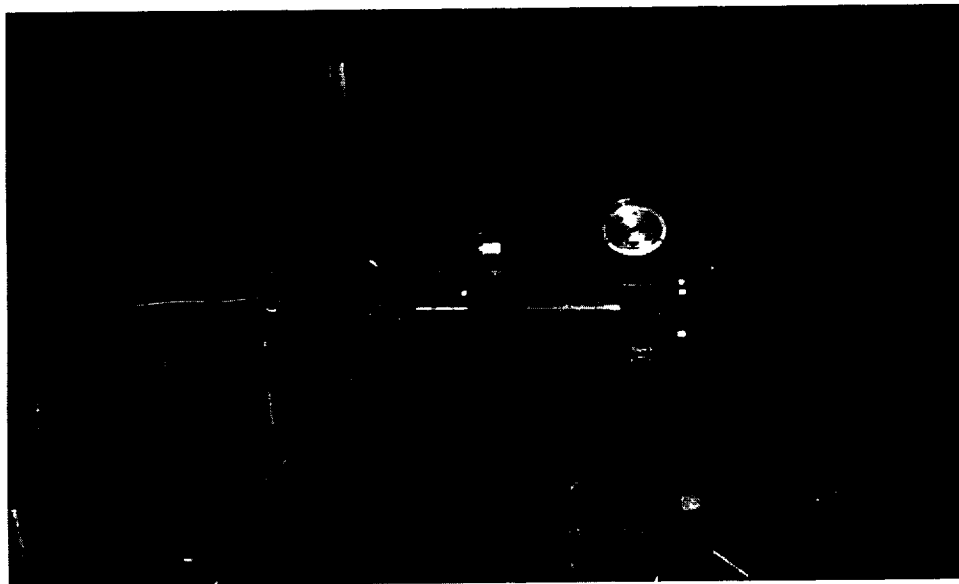


Figure 4.2 MR Damper Setup

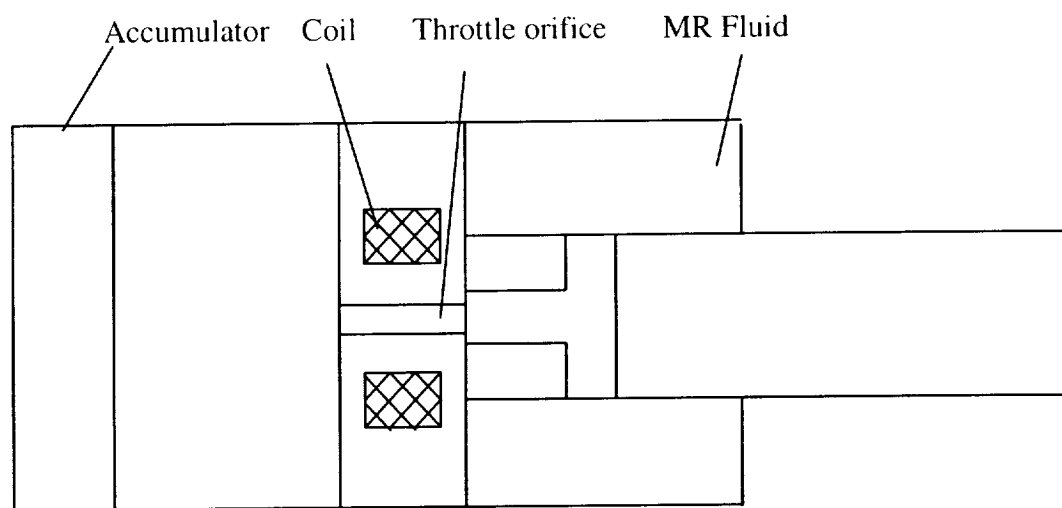


Figure 4.3 Principle Diagram of MR Damper

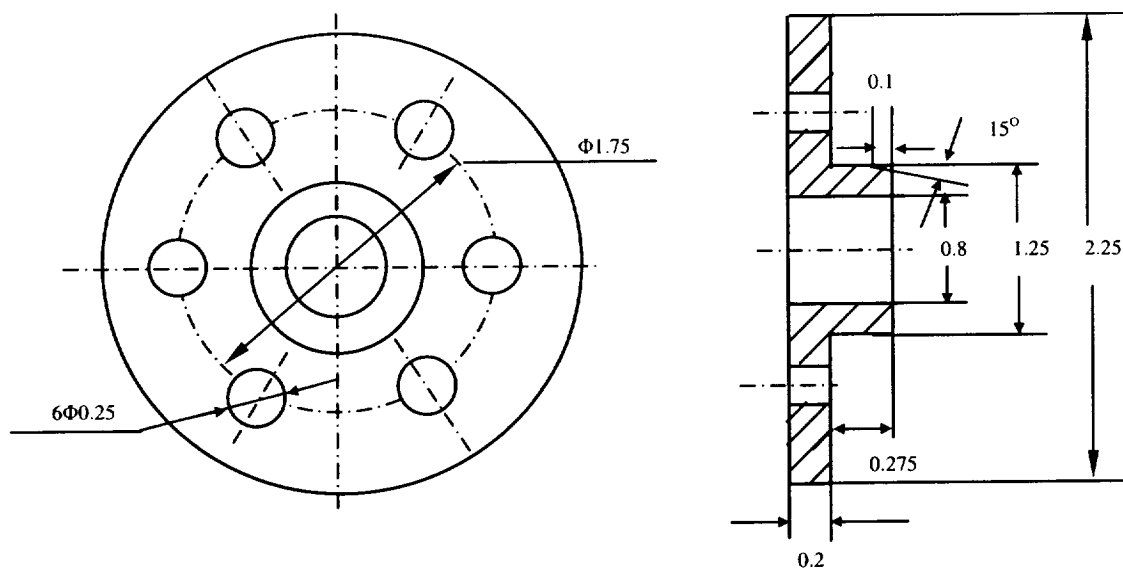


Figure 4.4 MR Damper Piston (Rear View)

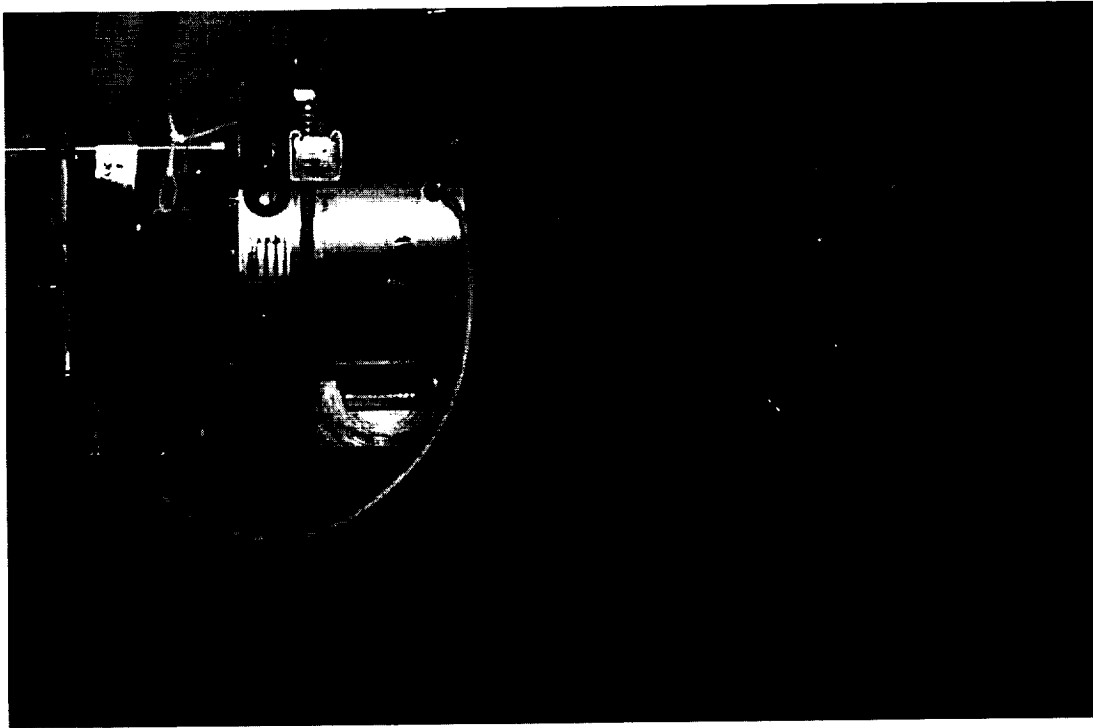


Figure 5.1 Shaker Setup

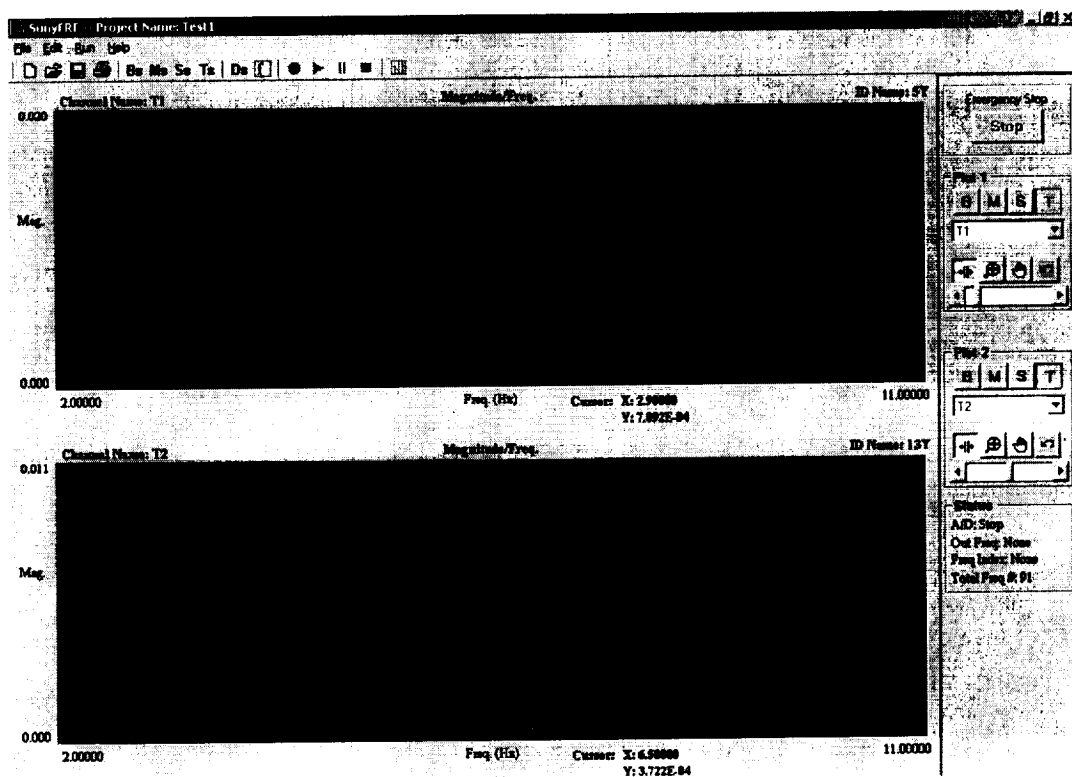


Figure 5.2 Screen of In-house Software

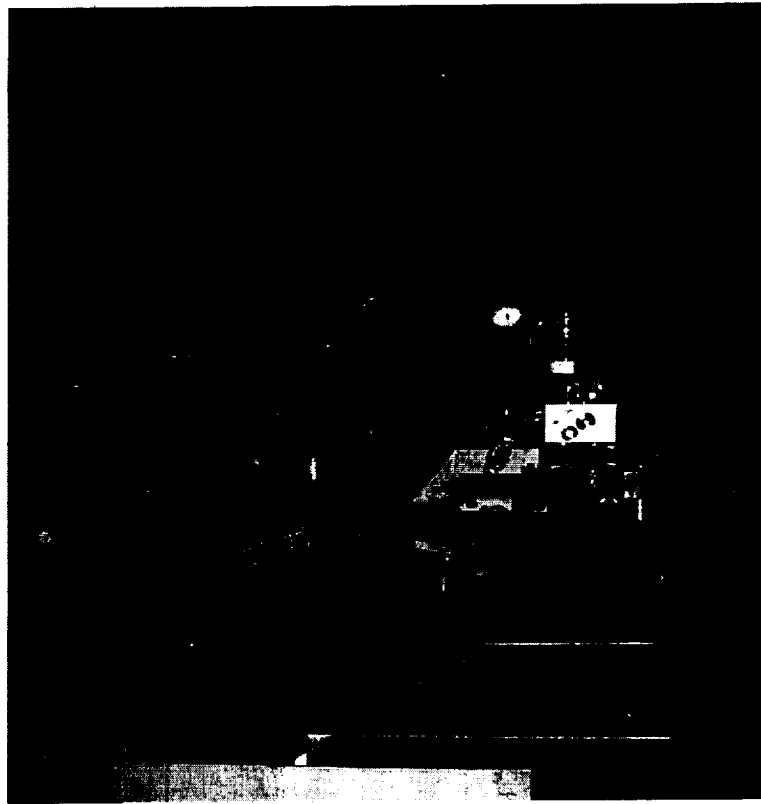


Figure 5.3 Experimental Setup

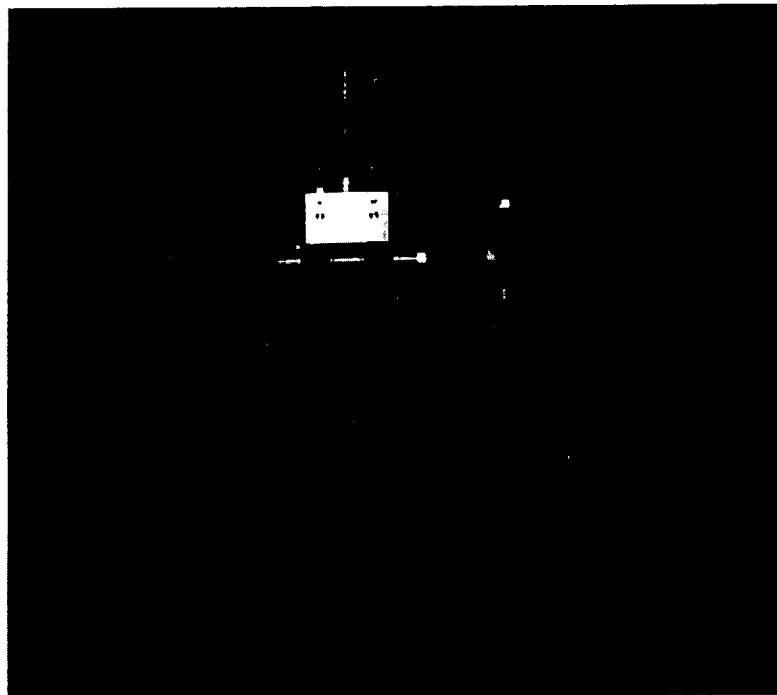


Figure 5.4 Damper Installation

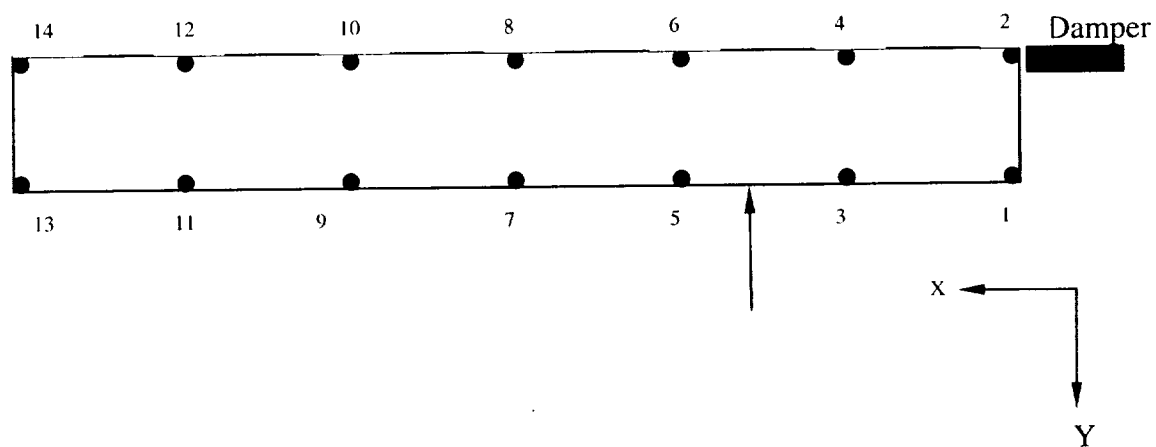


Figure 5.5 Node Layout

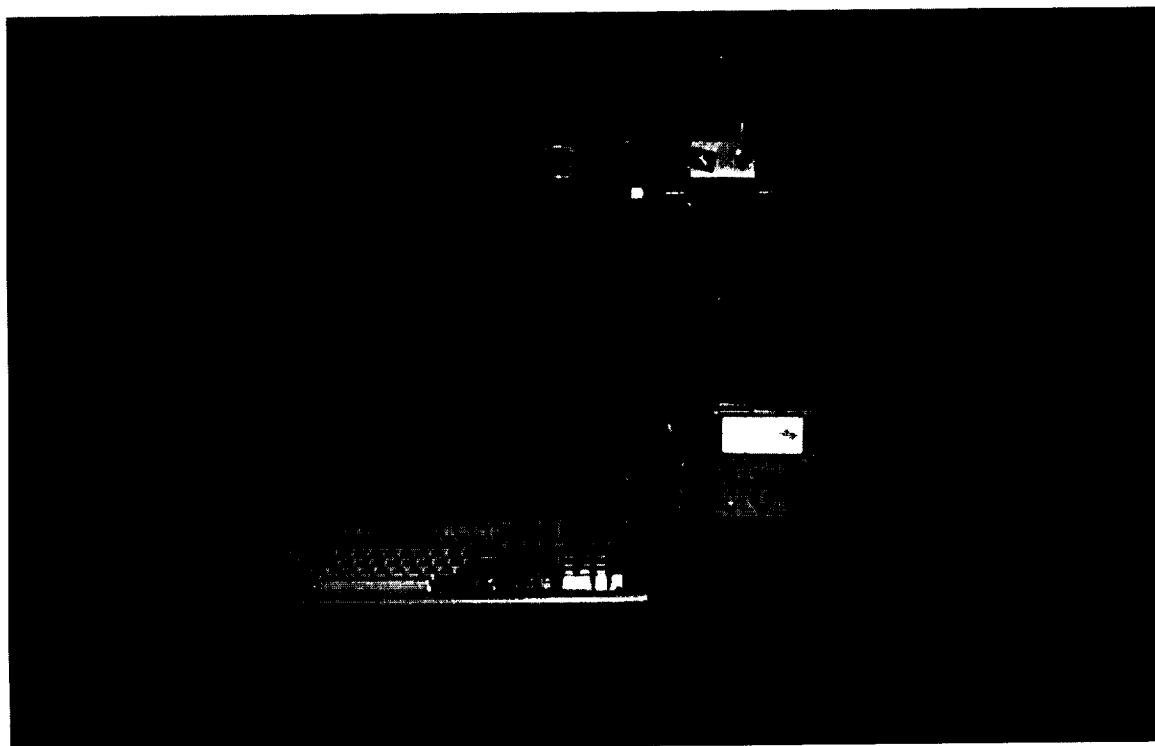


Figure 5.6 Controller Setup

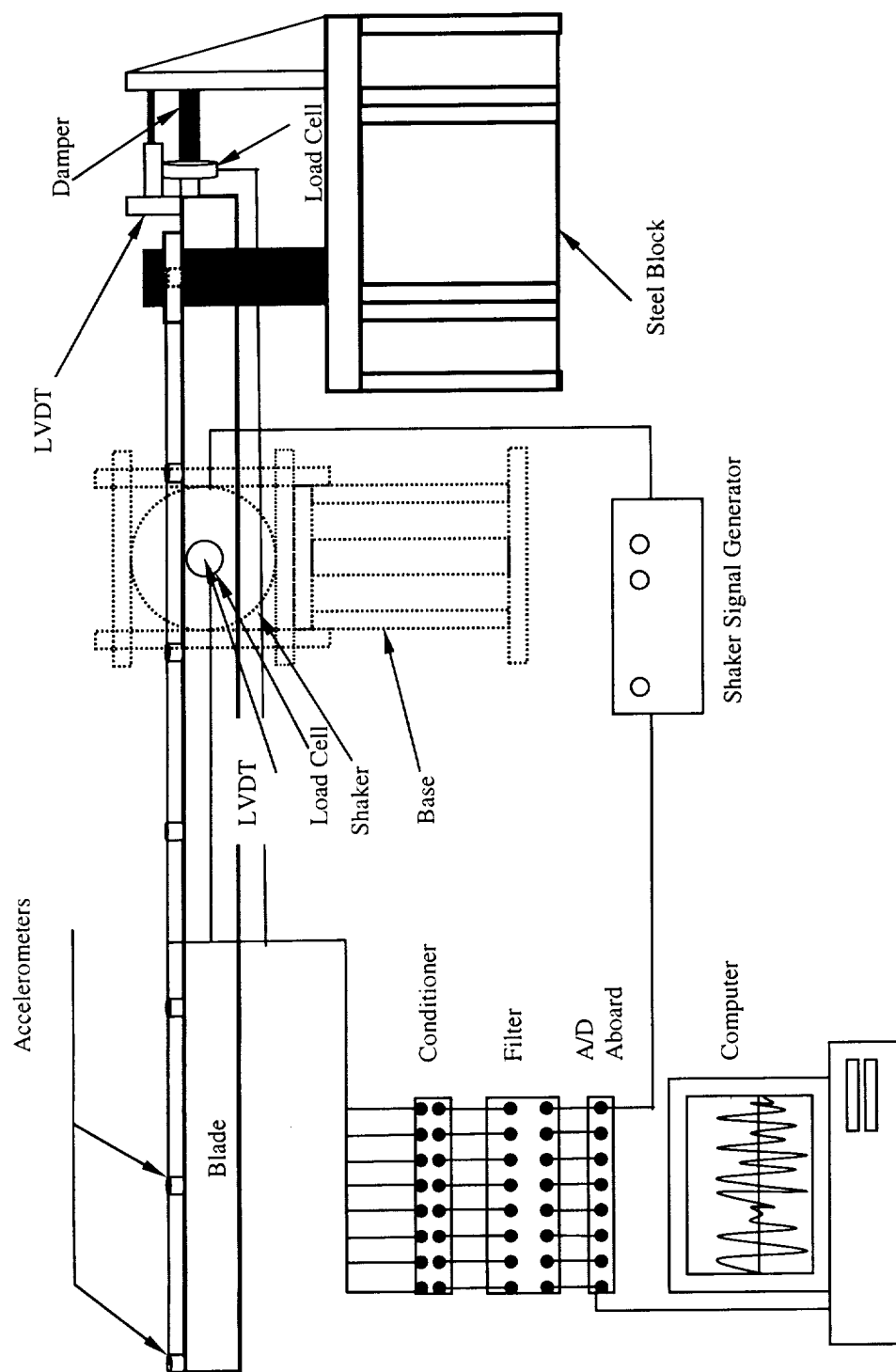


Figure 5.7 Data Acquisition System

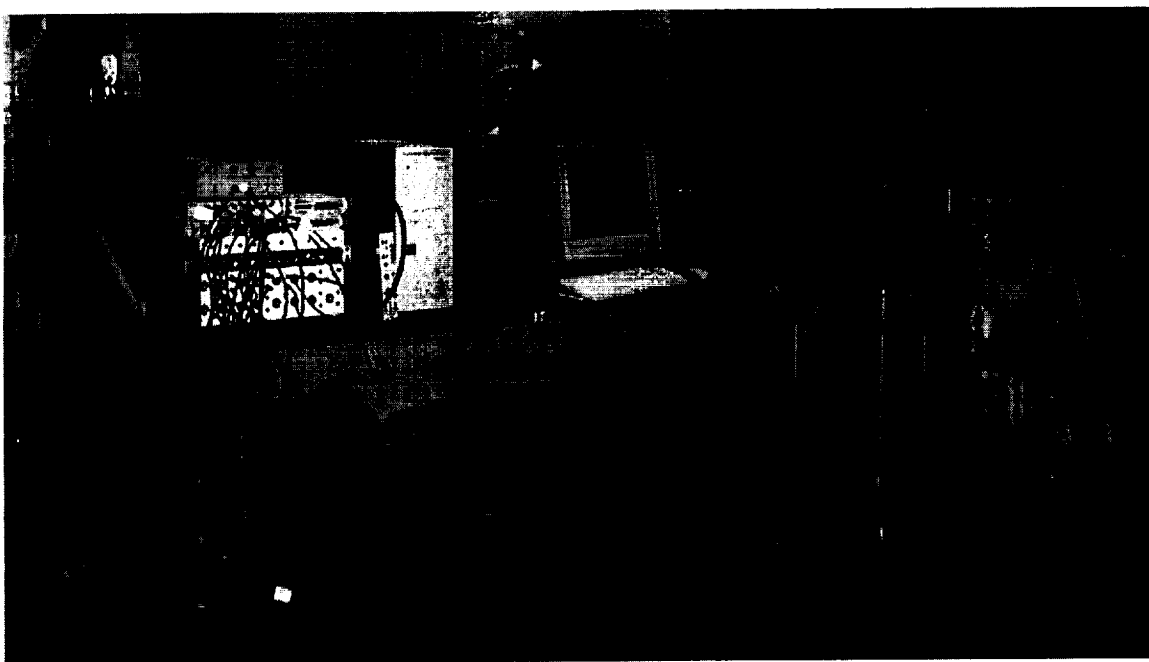


Figure 5.8 Data Acquisition System

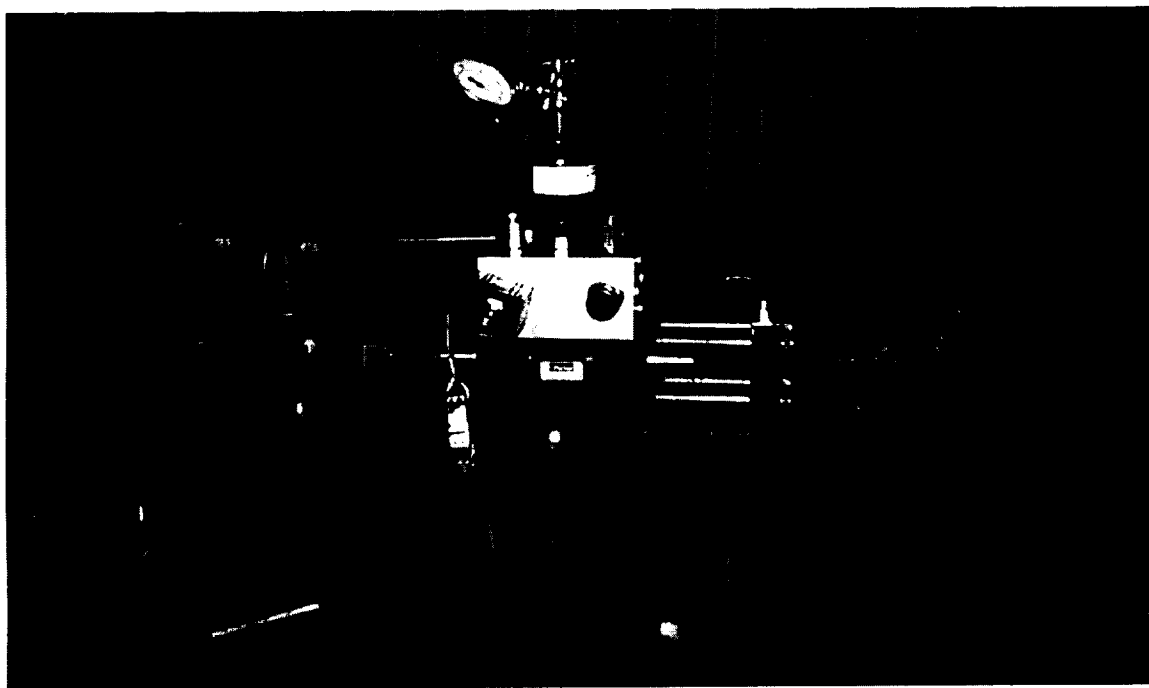


Figure 5.9 Damper Installation

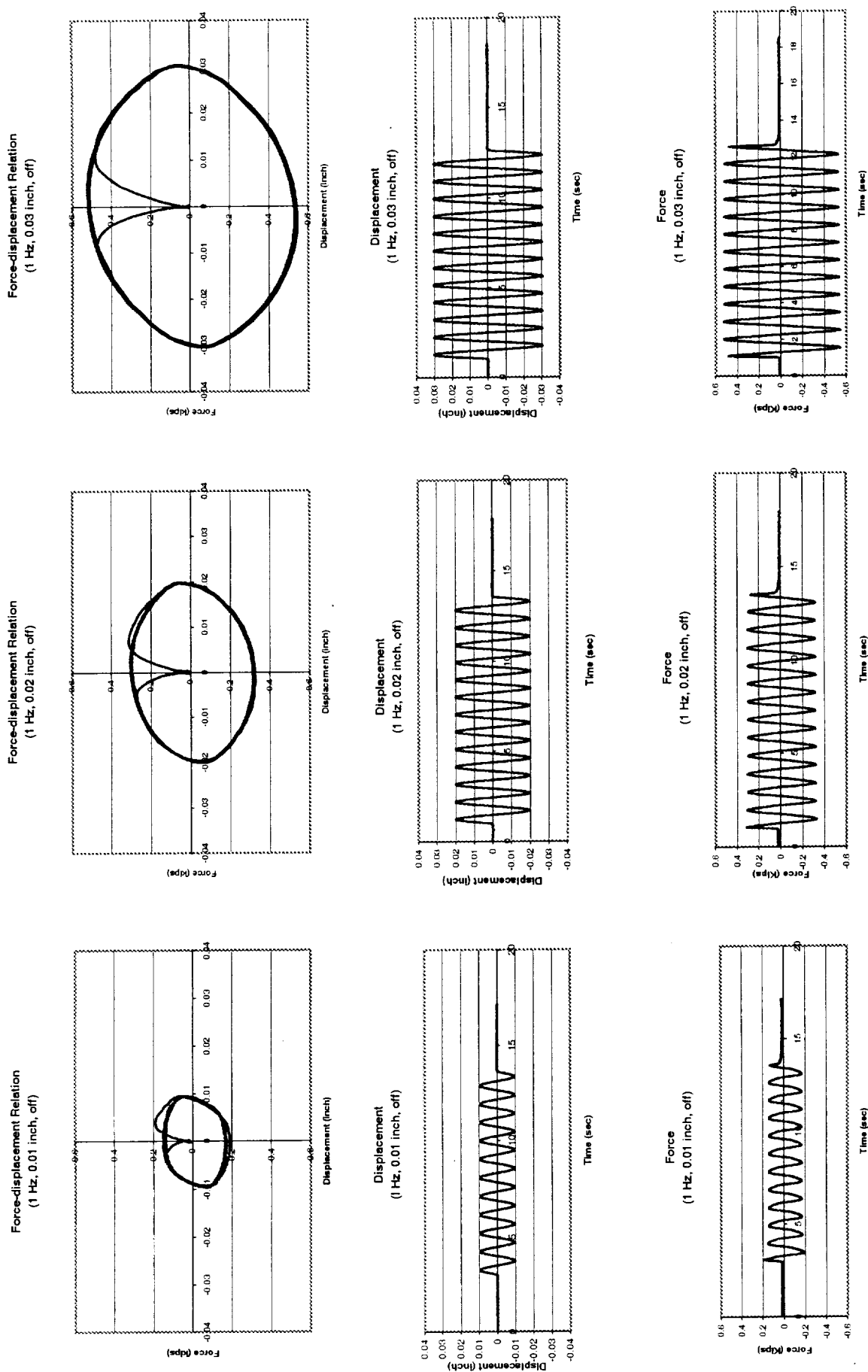


Figure 5.10 Force-displacement Relation (1 Hz, power off)

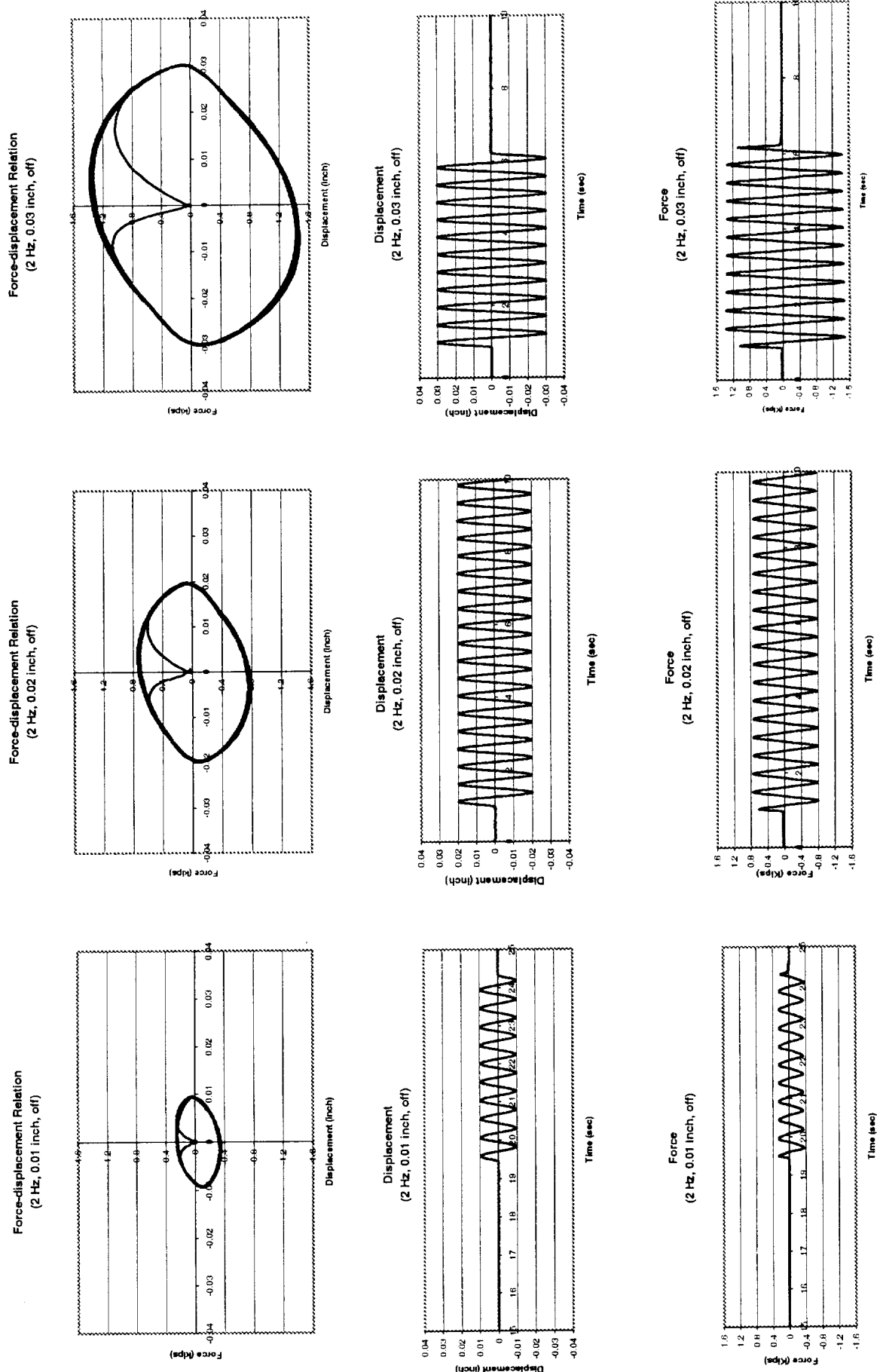
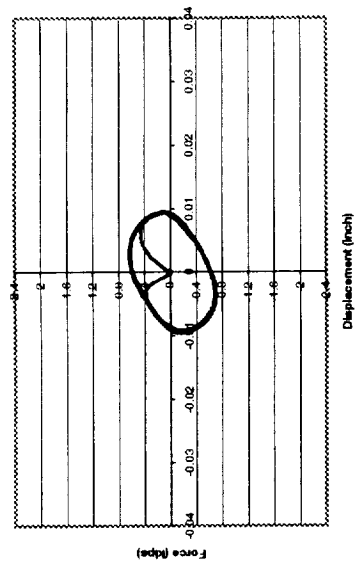
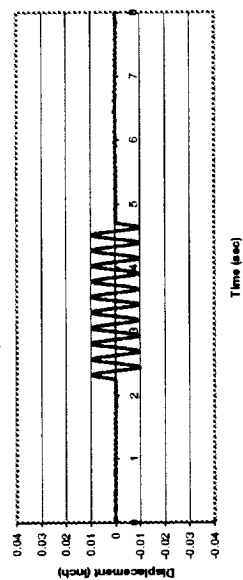


Figure 5.11 Force-displacement Relation (2 Hz, power off)

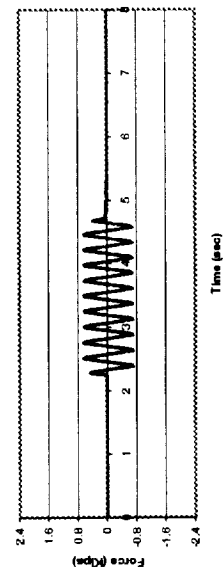
Force-displacement Relation
(4 Hz, 0.01 inch, off)



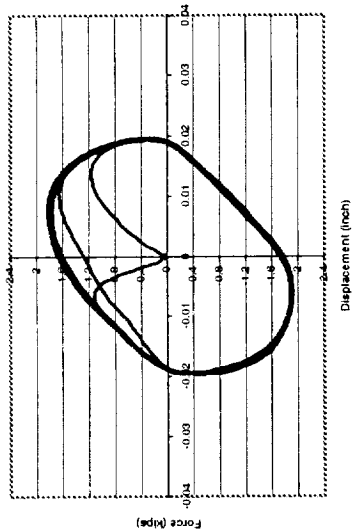
Displacement
(4 Hz, 0.01 inch, off)



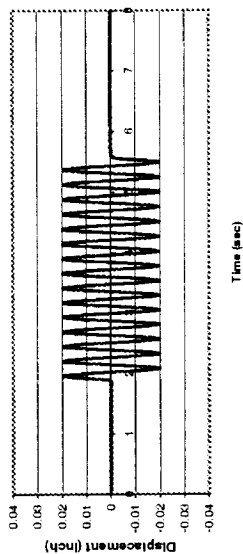
Force
(4 Hz, 0.01 inch, off)



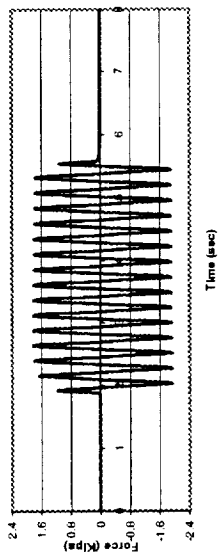
Force-displacement Relation
(4 Hz, 0.02 inch, off)



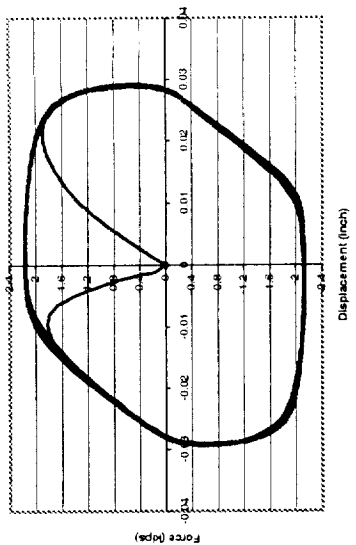
Displacement
(4 Hz, 0.02 inch, off)



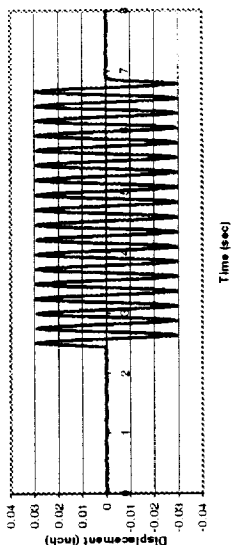
Force
(4 Hz, 0.02 inch, off)



Force-displacement Relation
(4 Hz, 0.03 inch, off)



Displacement
(4 Hz, 0.03 inch, off)



Force
(4 Hz, 0.03 inch, off)

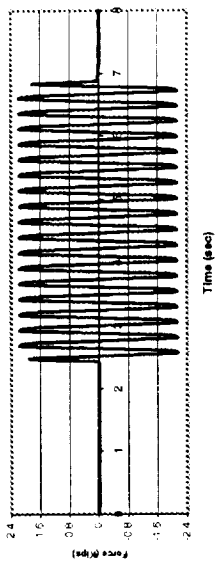
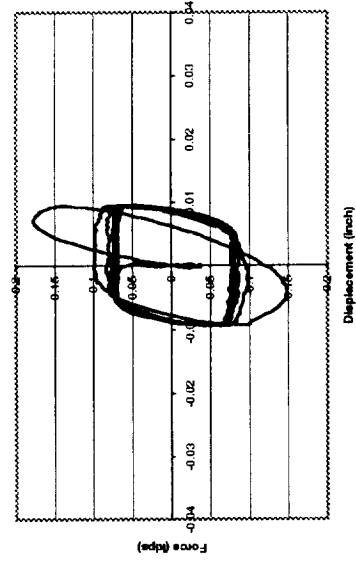
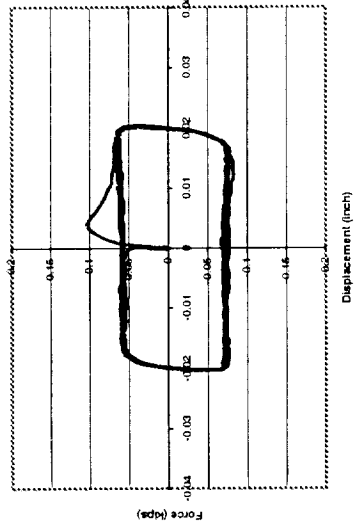


Figure 5.12 Force-displacement Relation (4 Hz, power off)

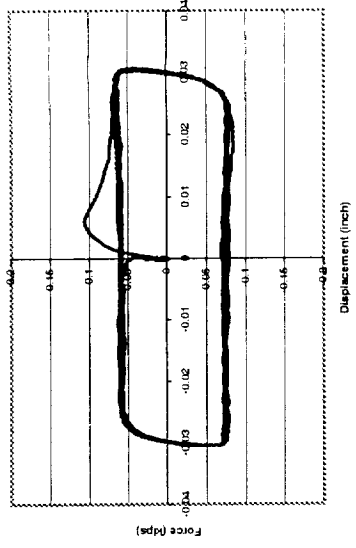
Force-displacement Relation
(1 Hz, 0.01 inch, on)



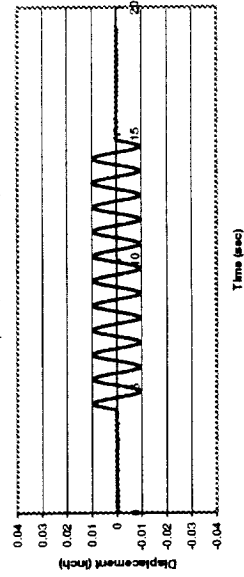
Force-displacement Relation
(1 Hz, 0.02 inch, on)



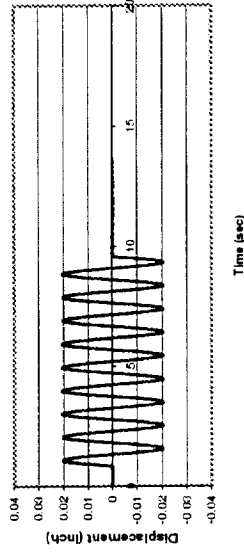
Force-displacement Relation
(1 Hz, 0.03 inch, on)



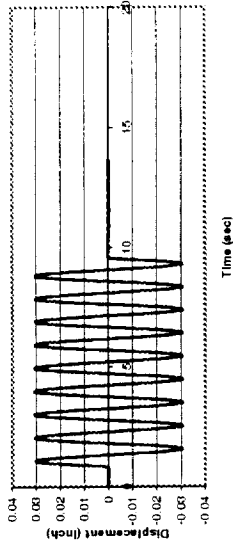
Displacement
(1 Hz, 0.01 inch, on)



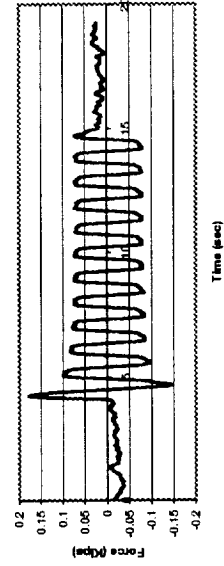
Displacement
(1 Hz, 0.02 inch, on)



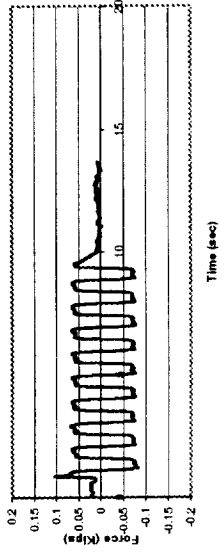
Displacement
(1 Hz, 0.03 inch, on)



Force
(1 Hz, 0.01 inch, on)



Force
(1 Hz, 0.02 inch, on)



Force
(1 Hz, 0.03 inch, on)

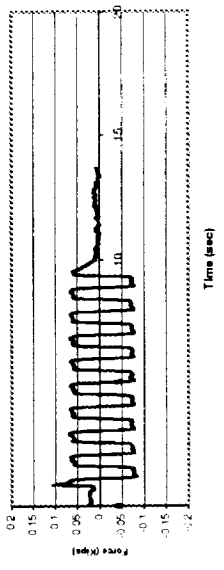


Figure 5.13 Force-displacement Relation (1 Hz, power on)

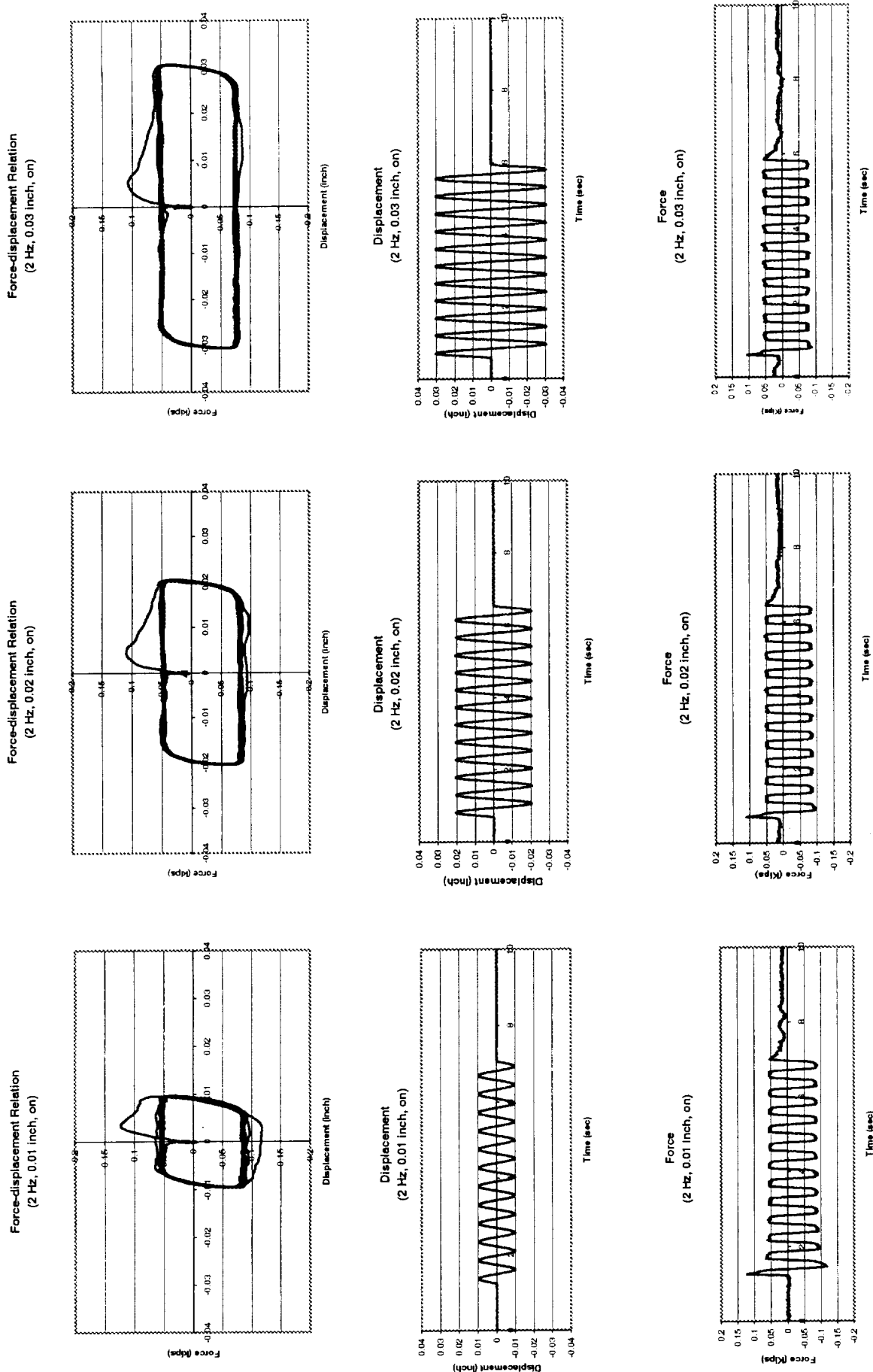
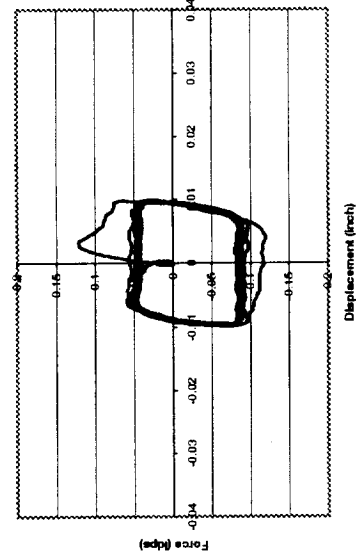
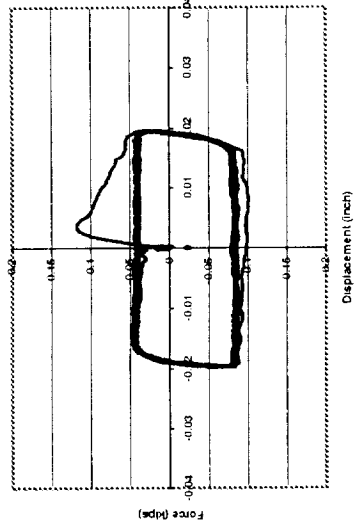


Figure 5.14 Force-displacement Relation (2 Hz, power on)

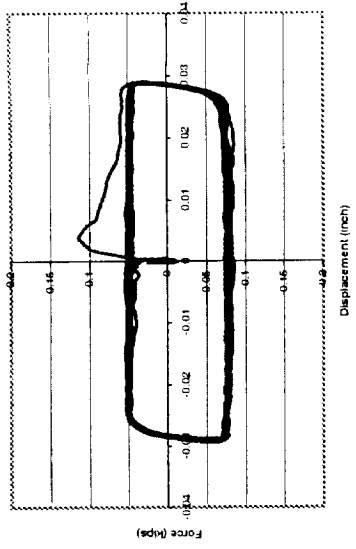
Force-displacement Relation
(4 Hz, 0.01 inch, on)



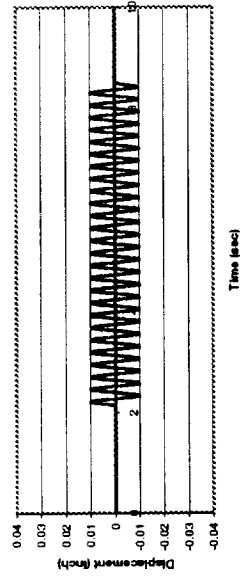
Force-displacement Relation
(4 Hz, 0.02 inch, on)



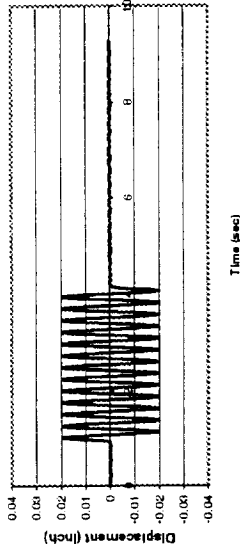
Force-displacement Relation
(4 Hz, 0.03 inch, on)



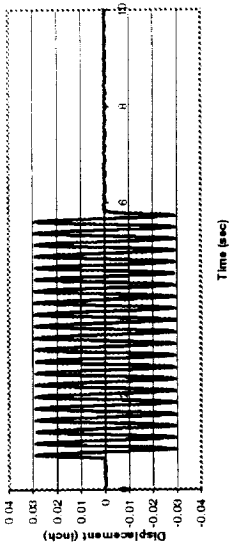
Displacement
(4 Hz, 0.01 inch, on)



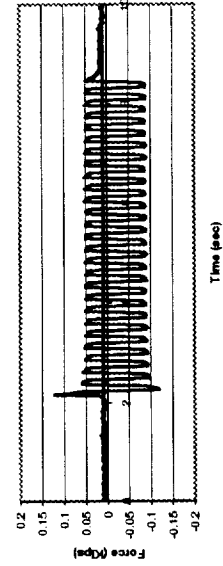
Displacement
(4 Hz, 0.02 inch, on)



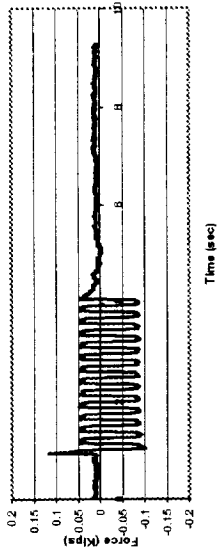
Displacement
(4 Hz, 0.03 inch, on)



Force
(4 Hz, 0.01 inch, on)



Force
(4 Hz, 0.02 inch, on)



Force
(4 Hz, 0.03 inch, on)

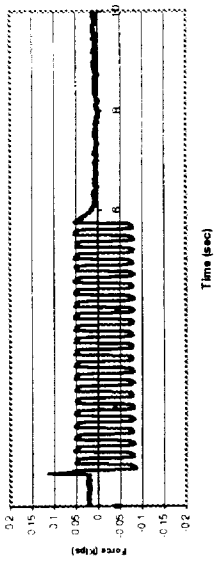
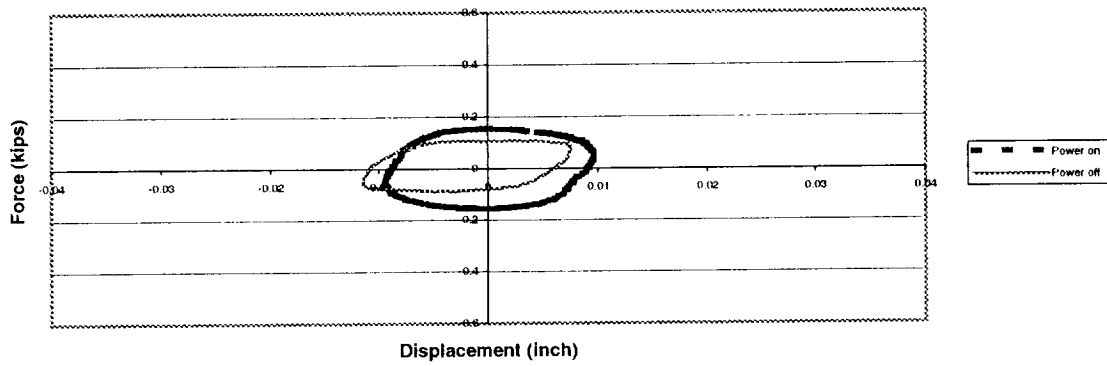
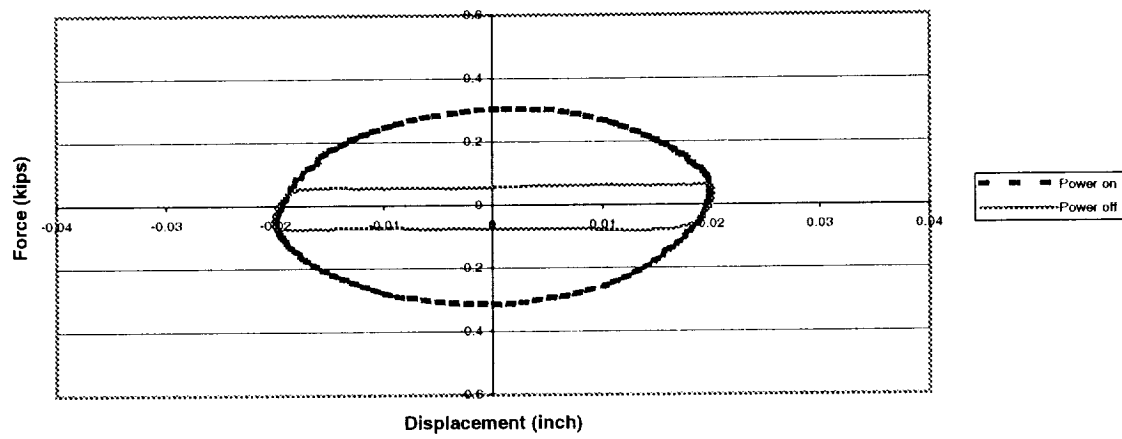


Figure 5.15 Force-displacement Relation (4 Hz, power on)

Force_displacement Relation
(1 Hz, 0.01 inch, Ratio=1.51)



Force_displacement Relation
(1 Hz, 0.02 inch, Ratio=3.56)



Force_displacement Relation
(1 Hz, 0.03 inch, Ratio=5.41)

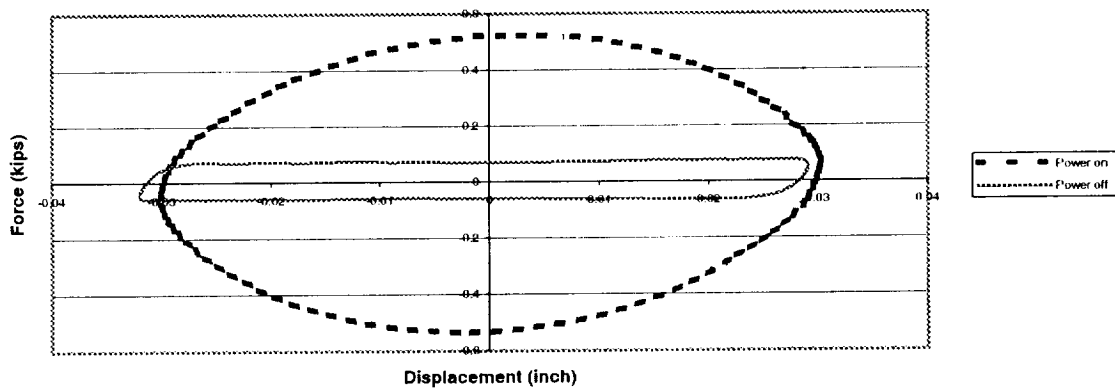
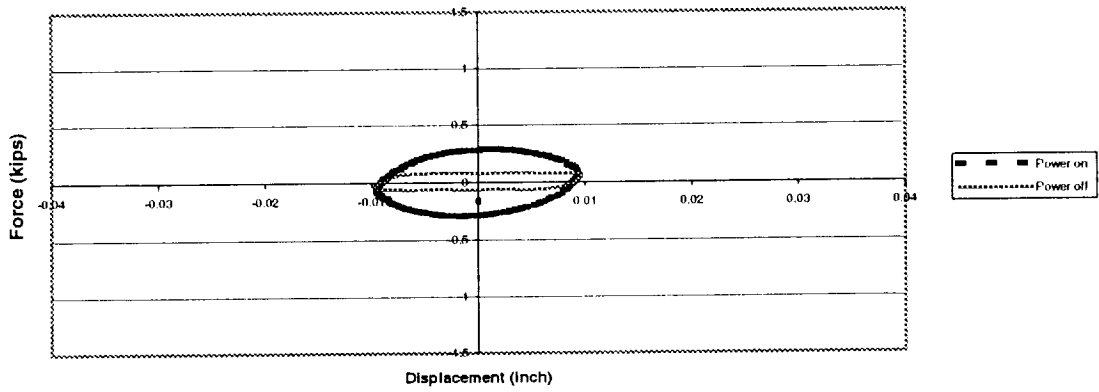
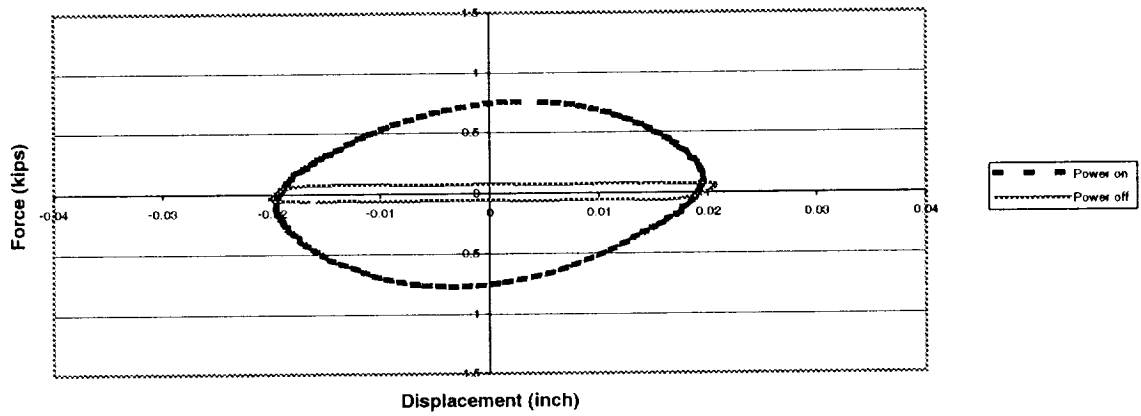


Figure 5.16 Comparison of the Force-displacement Relation (1 Hz)

Force_displacement Relation
(2 Hz, 0.01 inch, Ratio=3.34)



Force_displacement Relation
(2 Hz, 0.02 inch, Ratio=7.71)



Force_displacement Relation
(2 Hz, 0.03 inch, Ratio=16.08)

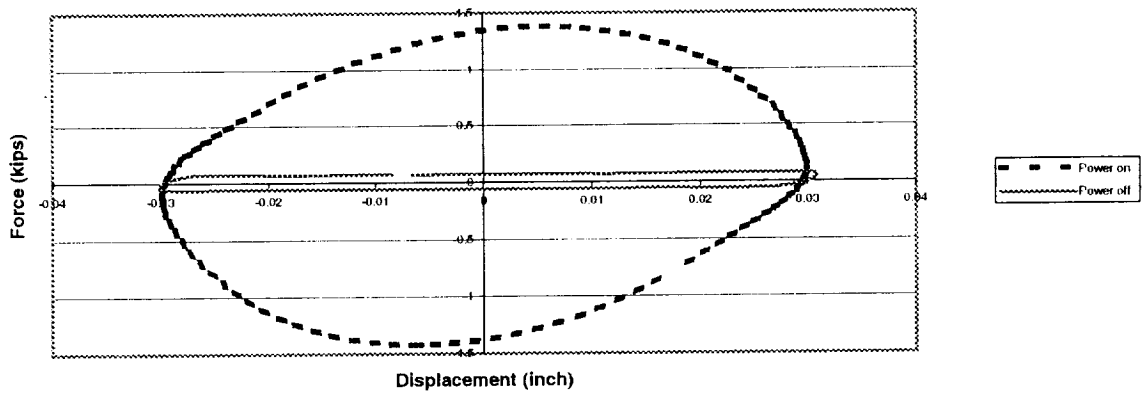


Figure 5.17 Comparison of the Force-displacement Relation (2 Hz)

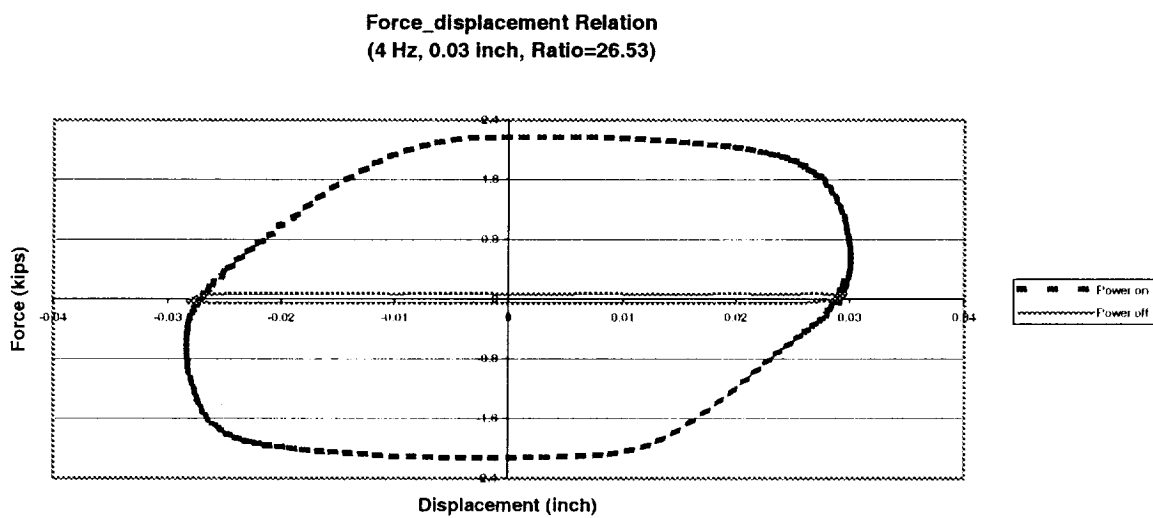
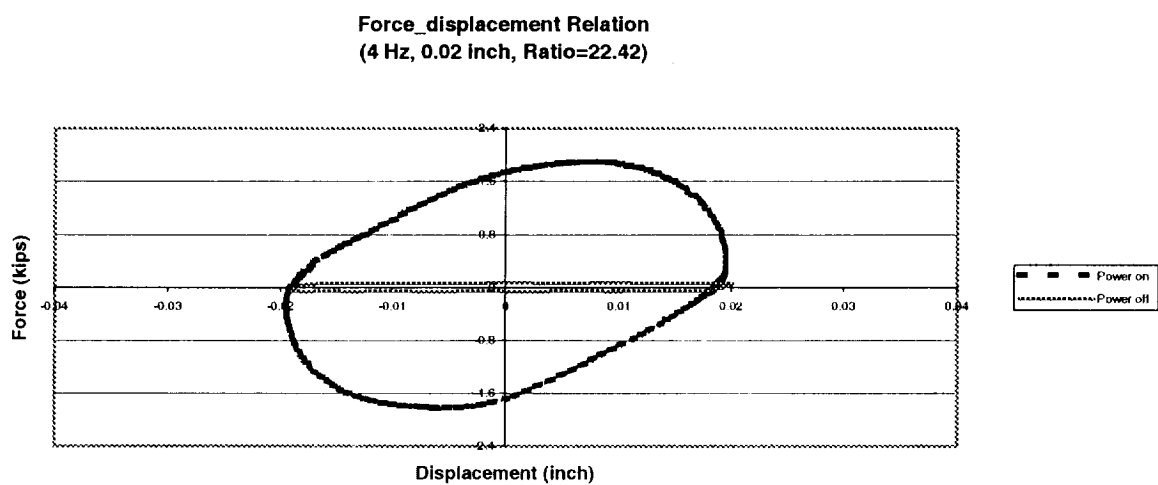
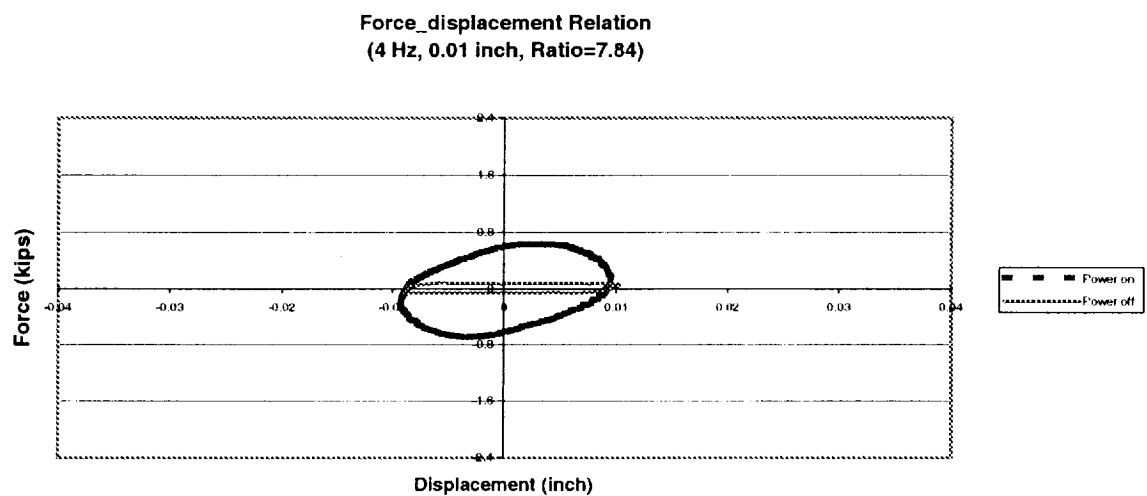


Figure 5.18 Comparison of the Force-displacement Relation (4 Hz)

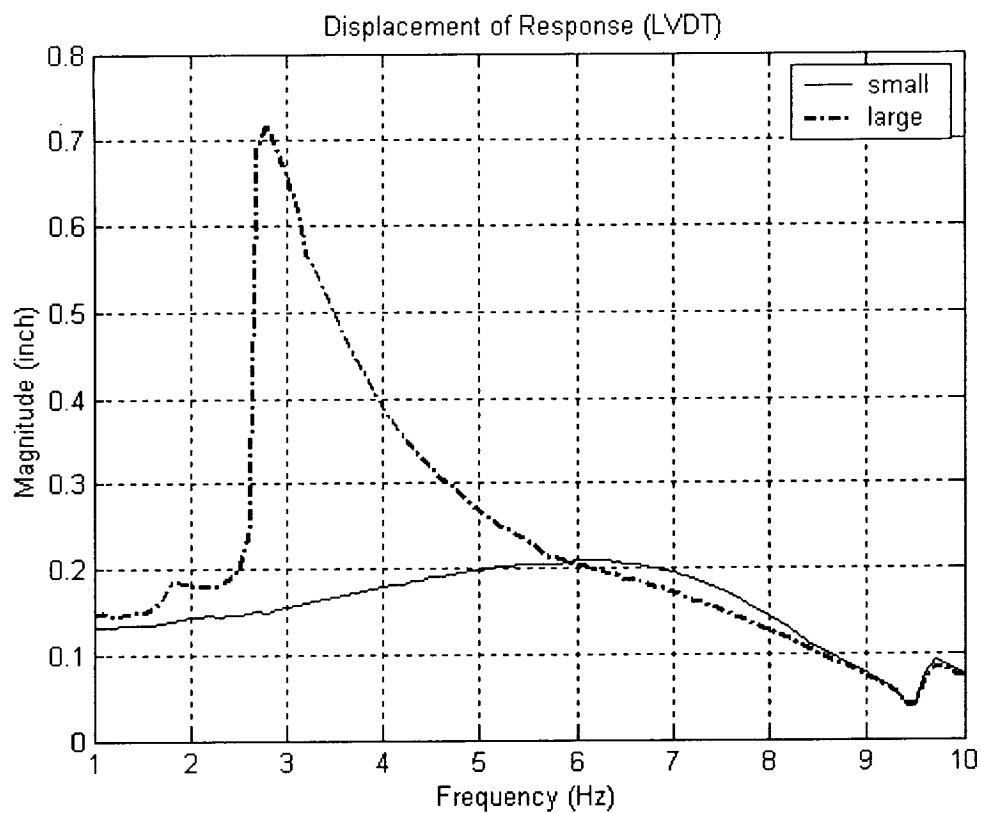


Figure 5.19 Displacement of Blade in the Two States

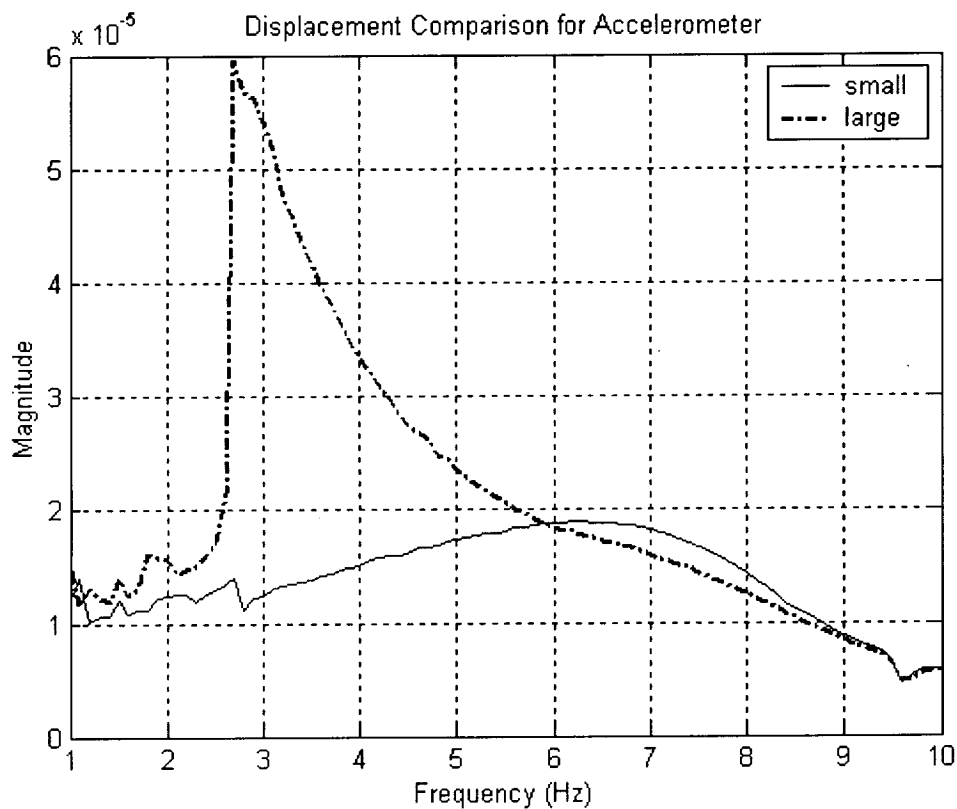


Figure 5.20 Displacement Comparisons for Accelerometer in the Two States

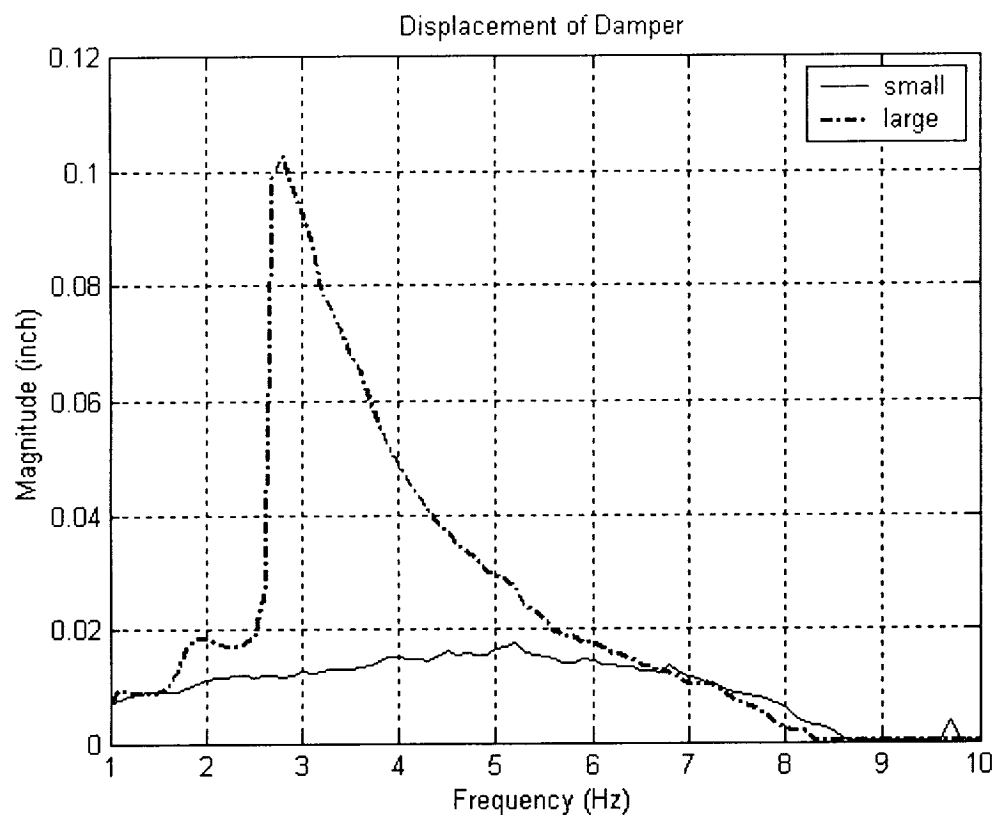


Figure 5.21 Displacement of Damper in the Two States

Displacement of Damper

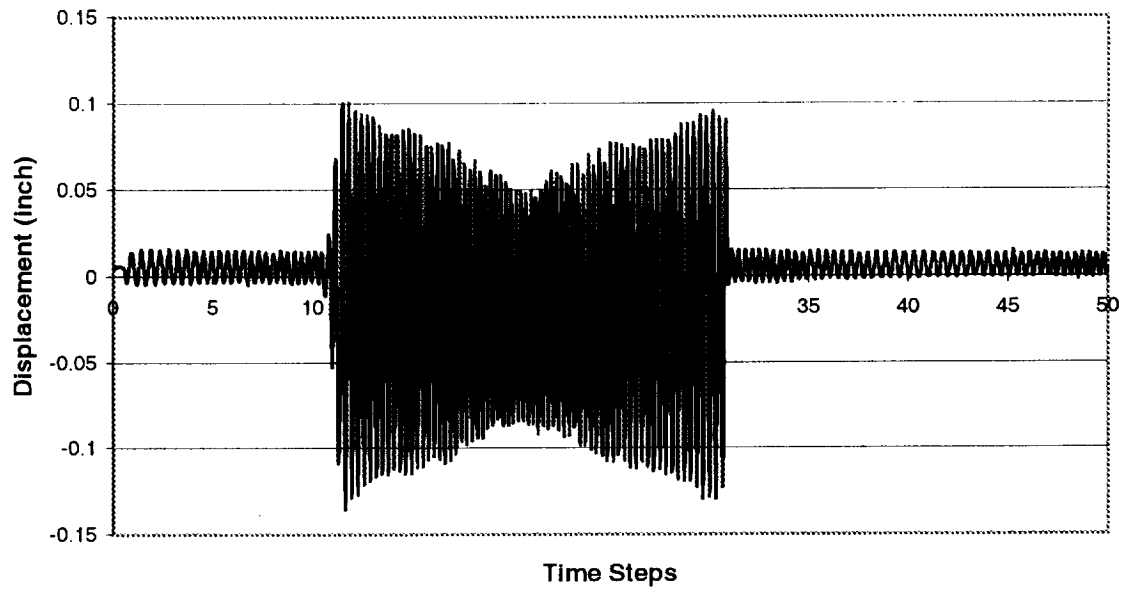


Figure 22 Displacement Comparison of Damper in Auto Sweeping Test

Displacement of Blade

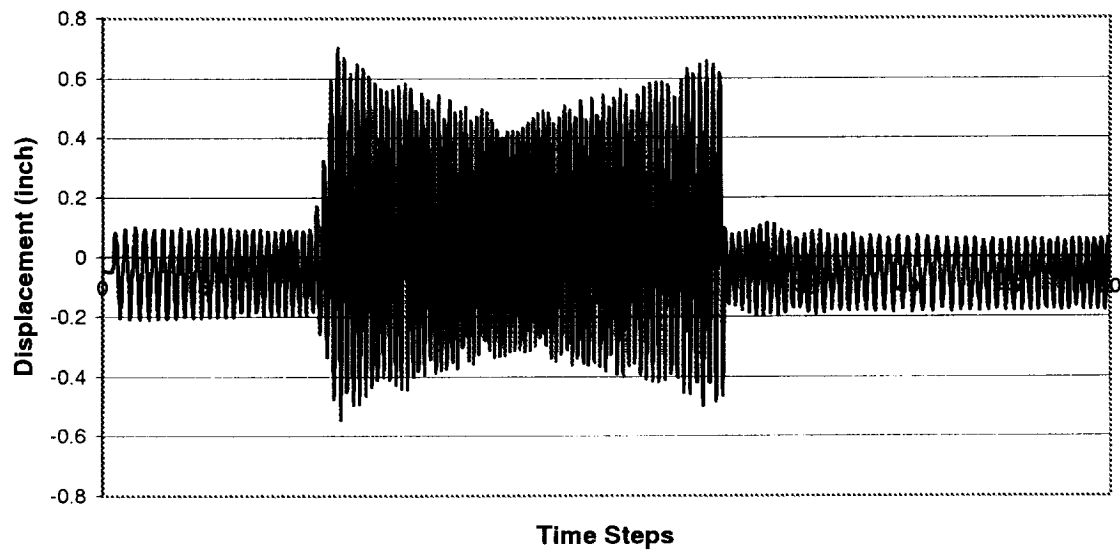


Figure 23 Displacement Comparison of Blade in Auto Sweeping Test

